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Fix the Responsibility

BEFORE the Legislature of the State of New York adjourns its present session drastic action should be taken to fix the responsibility for a condition which has become apparent in the building of the New York State Barge Canal. Although the canal and its locks are intended to accommodate barges capable of carrying 3,000 tons of cargo, yet at various points sharp bends in the canal will prevent the navigation of barges carrying more than 2,000 tons. In other words, the capacity of barges destined for use on the State Barge Canal must be one-third less than the locks of the canal are capable of accommodating.

The tendency in shipbuilding of late has been towards a steadily increasing size of cargo vessels. In fact, it is safe to say that the average cargo vessels of to-day have double the cargo capacity of similar vessels built a few years ago. As it is estimated that when the barge canal is completed the time of passage from Lake Erie to the Hudson River will be reduced from ten to four days, it surely is imperative from an economical standpoint that the largest possible vessels can be used in this service.

Would it not be well for the Legislature to have these sharp curves straightened out now while it is comparatively easy to do so, rather than to wait until the canal is in commission, and then have to do the work later at much greater expense?

Meantime, the authorities should find out who is responsible for these restrictions.

Classification and Filing of Engineering Data

TO obtain the greatest possible benefit from the mass of technical information and data published in the engineering press and in the proceedings of technical societies, the drafting departments of well-organized shipbuilding establishments usually adopt some system of classifying and filing such data so that it may be immediately available for reference. Considering the wide variety of sources from which such data are obtained, and the diversified forms in which it is presented, it is hardly likely that there is much uniformity in the reference files of different shipyards. The basis of the system in probably every case is, or should be, a card index arranged so that any portion of an article or paper can be instantly found, but the details may vary widely. As this subject is well worth a thorough discussion, we would suggest that our readers send to this office a brief statement explaining the methods which they are using for filing and classifying such engineering data. It will then be possible to present in a future issue a symposium in which methods can be compared and perhaps the best and most useful system established as a standard.

Status of the Submarine

IN the series of articles on submarines by Mr. Simon Lake, which is concluded in this issue, attention has been directed to the improvements which must be made in future submarines to make them effective weapons for offensive warfare. The status of the submarine as it exists to-day is such that it may very properly be called a perfected craft as far as its construction and methods of operation are concerned. No further improvements are essential or necessary in the principles of operation. Types of submarines are in existence whose capabilities are such that when proper attention has been given to stream lines, stability and pressure-resisting qualities, they may be designed for any desired service, so far as the boat itself is concerned. But the weakness of the vessel lies in its propulsive machinery, for both surface and submerged operation.

As far as prime movers for submarines are concerned, steam machinery has generally been found unsuited for the work, the principal objections to its use being the difficulty in instantly housing the comparatively large smokestack when submerging, the valuable space occupied, the unavoidable storage of heat in a confined space, and the impossibility of immediately getting underway at full power on coming to the surface after a long submergence. The gasoline (petrol) engine has been discarded because of the numerous explosions resulting from the accumulation of dangerous gases within the hull. The Diesel engine, while generally accepted at the present time as the motive power for submarines, has by no means proved entirely satisfactory. Although this type of engine has given excellent service in surface vessels, where the weight and space occupied are not important features, nevertheless, when it comes to an installation in a submarine, the size and weight of the engine are strictly limited, while at the same time a wide range of speeds is demanded. Reversibility is also a desirable feature, but it has been obtained in the Diesel engine only at the expense of much additional complication. For submerged work, the storage battery is excessively heavy for the energy which it stores. It is also bulky, so that only sufficient energy can be carried to propel a modern submarine while submerged at about 11 knots for one hour, at about 8 knots for three hours, or at about 5 knots for twenty hours.

While Mr. Lake emphasizes the necessity for immediate improvement in the propelling machinery of submarines if the vessel is to be successful for offensive warfare, he points out that the improvements are by no means so important if the submarine is to be restricted to defensive warfare only. For purely defensive purposes, the present types of submarines, notwithstanding their defects, have conclusively proved in the present war their capability of maintaining their lines of defense,

while in addition some of them have found time to do effective offensive work. As a strictly defensive weapon, Mr. Lake has laid special stress on the type of craft which he has termed the "amphibious submarine"—that is, a small vessel of about 200 tons displacement which can be hauled out of the water on special trucks and transported at high rates of speed over the railways on land to points of attack. Such vessels, Mr. Lake estimates, could probably be built for about $300,000 (£61,500) each, capable of making a speed of 10 knots on the surface and about 8 knots submerged. In the United States such small boats can be transported from New York to San Francisco in three days, from New York to Boston in five hours, or from New York to New Orleans in thirty-six hours in perfect safety.

Such a system of coast protection, enabling the quick mobilization of a large number of submarines at any threatened locality, while a distinct novelty, has much to recommend it from the standpoint of economy, utility and expediency.

Submergence Test of a Submarine

One of the interesting performances required of a submarine during its acceptance trials is that known officially as the deep submergence test. In most cases in the United States Navy the requirement for coast submarines is a submergence to a depth of 200 feet for a period of not less than ten minutes, the hull not to suffer any serious deflections nor develop any serious leaks during this time. The pressure at this depth in sea water is 88 pounds per square inch.

A test of this nature was recently successfully performed on the Lake type submarine *G*-3, recently completed at the New York Navy Yard. The test was carried out under the supervision of a naval constructor by a wrecking and salvage company, especially skilled in this kind of work.

A few days before the test was to be made a spot in Long Island Sound, at which there is approximately 200 feet depth of water, was located and buoyed. The vessel at the navy yard was overhauled, valves tested, depth gages calibrated, air flasks and batteries charged, salvage hoisting straps fitted around the hull, and all the various arrangements necessary to ensure a successful test were completed. As it was desired to complete the test in as short a time as possible, the vessel was put in condition, practically ready for her submergence, at the yard and then towed by a seagoing wrecking tug to the scene of the test. In addition to the tug, the contractor furnished a salvage vessel, which was equipped with air compressors and diving gear.

A preliminary submergence was made in about 60 feet of water to familiarize the men who conducted the test with their duties, and, in order to see that every detail was in perfect order. There are two practicable methods for submerging vessels on this kind of test—one is to lower the vessel with no crew on board. Slings are placed about the vessel and fastened to a ship's crane; a small amount of negative buoyancy is then taken and the ship sealed from the outside and lowered. This method has the disadvantage that no inspection can be made until the vessel comes to the surface, and that there is no guide, except the strain on the cable, as to whether the vessel is taking in water. The other method is to haul the vessel down on its own anchor cable with positive buoyancy. In this method a skeleton crew is aboard and complete control of the vessel is maintained. The vessel may be stopped at any depth and inspection made for leaks, deflections of hull taken, etc. In case of emergency the vessel can be brought to the surface immediately by blowing the ballast tanks with compressed air.

This latter method was employed for the *G*-3, a crew of four men with two observers going down with the vessel. The ship's anchor was lowered to the bottom and all hatches closed and secured. Then water ballast was taken on until the vessel had less buoyancy than the weight of the anchor. The ballast was so distributed as to give the vessel three to four degrees trim by the stern. Then, when the strain was put on the anchor cable the vessel settled to an even keel, thus avoiding any planing effect during descent.

Deflection battens were rigged at certain frames for measuring elongation and compression of the hull—observations were taken at depths of 50, 100, 150 and 200 feet. These data are not available for publication, but it may be stated that the vessel was found to be tight and the hull deflections not excessive.

Communication was maintained with the surface by the telephone, which is carried in a marker buoy as a part of the vessel's equipment. High-pressure air hose was attached to a valve in the ship's side, and the upper end fastened to the small boat which carried the telephone. In case of necessity, the air hose would have been connected to the salvage vessel's compressed-air line, and the *G*-3, thus filled with high-pressure air, which would have excluded any water and brought the vessel to the surface.

The vessel actually remained at the maximum depth about 14 minutes. On coming to the surface, the ballast tank valves were secured, the anchor hauled up and the vessel towed back to the navy yard—the total time of the test consuming less than 24 hours.

Form Coefficients of Vessels

BY ARTHUR R. LIDDELL

The various coefficients of form of a ship are usually treated as arbitrary factors, to be varied according to the taste of the designer. In reality they are much less arbitrary than they seem, and the designer's taste in the choice of them varies between not very wide limits. It is true that the block coefficients of cargo steamers may range somewhat from .6 to .85, but the lower limit is met with only in very small and the higher one only in very large vessels.

If we consider the prismatic coefficient, *i. e.*, that resulting from the division of the displacement by the product of midship area multiplied by length, as the deciding one for comparative fineness of ends of vessels of given dimensions, we find that a degree of fineness of .65 in a 100-foot ship will correspond with one of about ½ (1 + .65) = .825 in a 400-foot ship and with one of about 1/3 (2 + .65) = .883 in a 900-foot ship; that is to say, with similar fineness of ends the 100-foot ship has no parallel middle body, the 400-foot one has a parallel middle body extending over half her length, and the 900-foot one has a parallel middle body extending over two-thirds of her length. The reason of this is that the proportions borne by the length to the breadth, to the draft, and to the square root of the midship area, respectively, increase with the size of vessels. Midship area in fact increases very much as length, or we may put it that $\sqrt{\text{midship area}}$ increases as $\sqrt{\text{length}}$ instead of directly as the length, as might at first sight be supposed. If we take designer's taste to vary by about 2 percent in the 900-foot ship, we may get .9 for her prismatic coefficient instead of .883, and the corresponding prismatic coefficient for the 100-foot ship becomes .7 instead of .65, giving a difference of about 8 percent. Let us take

TABLE OF COEFFICIENTS OF FINENESS A (FOR CARGO VESSELS)

Length in Feet.	Breadth in Feet.	Ballast Draft in Feet.	$\sqrt{\text{Length}}$ in Feet.	Prismatic Coefficient Φ	Block Coefficient δ	Loadline Coefficient α	Midship Section Coefficient β
100	22.3	11.2	10	.6 — .7	.51—.62	.71—.79	.84 —.89
225	33.4	16.8	15	.73— .8	.655—.75	.815—.865	.90 —.93
400	44.6	22.4	20	.80—.85	.75 —.81	.87 —.90	.93 —.95
625	55.7	28.0	25	.84—.88	.80 —.835	.895—.915	.95 —.955
(900)	(66.9)	(33.6)	30	.87—.90	.825—.865	.91 —.93	.955—.965
				Ratios in $\alpha^{1.5}$	which above coefficients vary: α^2	$\Phi^{\frac{2}{3}}$	$\sqrt{\alpha}$

TABLE OF COEFFICIENTS OF FINENESS B (FOR CARGO-PASSENGER VESSELS)

Length in Feet.	Breadth in Aeet.	Ballast Draft in Feet.	$\sqrt{\text{Length}}$ in Feet.	Prismatic Coefficient Φ	Block Coefficient δ	Loadline Coefficient α	Midship Section Coefficient β
100	20	10	10	.....	.....	.....	.....
225	30	15	15	.43 —.50	.33 —.40	.57—.63	.76—.80
400	40	20	20	.575—.625	.48— .535	.69—.73	.83—.86
625	50	25	25	.66 —.70	.58— .63	.76—.79	.87—.89
(900)	60	30	30	.72 —.75	.645—.69	.80—.83	.90—.91
				Ratios in $\alpha^{1.5}$	which above coefficients vary α^2	$\Phi^{\frac{2}{3}}$	$\sqrt{\alpha}$

$$\frac{\text{length}}{\text{breadth} \times \text{draft}} = \frac{1}{2.5}$$ as a standard for a cargo-passenger ship,

and

$$\frac{\text{length}}{\text{breadth} \times \text{draft}} = \frac{1}{2}$$ as a standard for a cargo ship,

and consider the corresponding tables *A* and *B*. These embody an attempt to reduce to order the variation which the element of size imposes on the coefficients of vessels. In these, it will be observed, the prismatic coefficient is made to vary with $\alpha^{1.5}$, the block coefficient with α^2, and the midship section coefficient with $\sqrt{\alpha}$, α being the coefficient of the load waterline. The coefficients given will be found to agree fairly well with general practice. Should the midship section coefficient chosen for a design be greater or less than that given in the table, the block coefficient in the same line may be increased or diminished to the same degree. The prismatic coefficient, *i. e.*, the degree of fineness of the ends of the vessel, then remains as before. It may be observed that the coefficient of the deckline will generally be about equal to the square root of the prismatic coefficient.

When a vesseel is lengthened, a parallel middle body having a block coefficient equal to the coefficient of the immersed midship section is inserted. The new block coefficient then becomes

$$= \frac{\text{old length} \times \delta + \text{additional length} \times \beta}{\text{new length}},$$

where δ is the old block coefficient and β the midship section coefficient. As regards speed, the lengthening of a vessel may be advantageous in case she has hitherto been too short for her area of midship section and for her speed. If her length hitherto, say for passenger work, has been = ½ (breadth × ballast draft), and the proportion borne by her length in feet to the product of $(\text{knots})^2 \times .563$ has been equal to 2¼, it is probable that any considerable increase of length will result in diminution of propulsive efficiency, in wetness, and in bad steering, while the carrying capacity will be less than if the midship section coefficient could have been suitably increased, as it would have been in a new design embodying the extra length.

The inner reason why, with given fineness of ends, the midship section of a small vessel is made finer than that of a larger one is that the critical wave, *i. e.*,, the wave having the same length as the vessel, becomes hollower with decrease of dimensions. To obtain good sea qualities, the rise of floor of a small vessel is, in ordinary practice, increased perhaps so far that an OG-bottom results, the base line being thereby artificially brought down farther below the main body. Were it not undesirable in view of docking and on other such grounds, the keel of a small steamer might be arched from sternpost to forefoot, and her floor amidships be designed with a very small rise, while the area of the midship section would be the same as that usually given.

The dimensions given in the tables are length, breadth and ballast draft, the later being made to vary with $\sqrt{\text{length}}$, and being regarded as the shallowest admissible for sea work.

The values given are not necessarily the best for the respective ranges of dimensions, but they agree fairly well with current practice. For drawing-office use they may be varied to suit the practice of the designer who may apply them, and any desired number of additional grades may be interpolated. The use of such tables will save a great deal of trial and error work. Assume, for instance, that a designer has particulars of a successful cargo vessel and wishes to obtain correspondingly good results in a larger one. A reference to Table A will show the necessary alteration in the proportions of length to area of midship section, etc., and the degree in which the coefficients of the smaller vessel must be varied to adapt them to the larger one.

Marked Increase in Japanese Shipping to the United States.—According to statistics published by the Department of Commerce at Washington for January, there were marked increases in the value of merchandise imported to the United States in American, British and Japanese ships. In January, 1916, as compared with January, 1915, American shipping increased about 60 percent, British shipping about 80 percent and Japanese shipping about 250 percent. For the seven months' period ending in January, 1916, as compared with the similar period ending in January, 1915, there was a gain of 55 percent for imports under the American flag, 10 percent under the British flag and about 125 percent under the Japanese flag. These figures show that while the war has stimulated trade under the American flag, the greatest benefits have been derived by British and Japanese shipping.

A Criterion of Propeller Surface

BY PETER DOIG

In solving the problem of best dimensions for a screw propeller to meet certain conditions, the three chief characteristics to be determined are diameter, pitch and surface; the thickness of blades, their number and outline and sectional shape being usually of secondary importance. When the power, speed and revolutions are of moderate amount, the order in which they may be placed is, first, diameter; second, pitch; and third, area of blades.

But when high power, speed and revolutions are under consideration the relative order then frequently becomes, first, area of blades; second, diameter; and third, pitch.

Assigning suitable diameters and pitches, though never an easy matter, is much assisted by deduction from experiment on model screws; but here little or no help is got for the computation of the area of blades which shall be sufficient and no more than sufficient; since it is well established by model and full-size experiments that efficiency is lessened, through frictional losses, by increase in area after a certain amount has been provided.

In the best modern practice the minimum ratios of developed surface to area of the disk enveloping the blade tips lie usually between about .28 and .34. Anything smaller, though indicating high efficiency in model experiments with very thin blades, involves in full-size abnormal root thickness to meet requirements of strength, and then returns relatively poor efficiency.

Large increase over these values is usually due to the desirability of providing sufficient breadth of blade to ensure effective contact with the water, or to diminish the intensity of thrust pressures; failure to guard against the alternatives being regarded as responsible for the phenomenon of "cavitation." By some investigators high blade velocities are thought to produce a certain "cascading" of the water round the blade resulting in parts of the surface giving no useful thrust, which defect has to be met by widening the blade; while by others the high thrust pressures per unit surface in screws of insufficient area are regarded as causing the pressure head, due to superincumbent water and atmosphere, to fail in supplying the screw with a "solid" medium.

High blade velocities are, *cæteris paribus,* proportional to revolutions, and similarly to diameter; and thrusts per unit surface increase with the value

$$\left(\frac{H E}{S D^2}\right)$$

where H is horsepower of engines per screw, E efficiency of screw, S speed in knots (HE/S being thus a measure of total thrust) and D diameter. Since these are the main factors taken into consideration by designers, there would seem a possibility of constructing some eclectic semi-empirical formula such as

$$\text{Disk area ratio of practice} = C + K R^m D^n \left(\frac{H E}{S D^2}\right)^p \ldots (1)$$

where C is the minimum disk area ratios of practice having the values previously mentioned (.28 to .34), and m, n and p are indices, giving approximation to the relative "weights" of R, D, and

$$\left(\frac{H E}{S D^2}\right)$$

in the expression.

On trial of various combinations of values being made with an extensive set of modern data, close accordance with practice was obtained with

$$K = \frac{1}{10{,}000}; \; m = 1; \; n = 3/4; \; p = 1/4.$$

Substituting in (1) this becomes

$$D.\ A.\ R.\ \text{of practice} = C + \frac{R D^{3/4} H^{1/4} E^{1/4}}{10{,}000\, S^{1/4} D^{1/2}} = C + \frac{R}{10{,}000} \sqrt[4]{\frac{D H E}{S}}$$

Since E varies between, say, .60 and .70, $\sqrt[4]{E}$ varies only between .88 and .91, and this term having but a small effect on the values of the formula may be neglected. We thus get:

$$D.\ A.\ R.\ \text{of practice} = C + \frac{R}{10{,}000} \sqrt[4]{\frac{D H}{S}}$$

It will be noted that mechanical efficiency of engine and variation of "wake" are not considered in the formula. But as they also enter as their fourth roots they are, like E, of minor effect.

The values of this expression, using .30 for C throughout, have been calculated for the results of over forty vessels, most of which will be found in the technical press*, and tabulated side by side with the actual area ratios. None of the fast vessels in this list has been a failure through cavitation; good performances as far as propulsion is concerned having been obtained in all cases at the *designed* speed.

On comparing the two columns of disk area ratios, it will be observed that higher values than the actual are generally given by the formula in the case of destroyers. This accords with the practice of other navies than the United States (to which the examples given belong); and, further, allows for the extent of "pressing" above designed speeds, to which they have been subjected. For example, No. 18, the *Monaghan,* at the designed speed of 29.5 knots, would take about 11,200 brake horsepower at 770 revolutions. This gives

$$D.\ A.\ R. = .30 + \frac{770}{10{,}000} \sqrt[4]{\frac{5.25 \times 3730}{29.5}} = .69$$

This is very close to the actual value, which was .68. Then again, in No. 15, the *Drayton,* an explanation of the large value obtained by the formula will be found in the low propulsive efficiency, .43, which immediately suggests cavitation and insufficient area for the "pressed" speed.

In the case of reciprocating-engined vessels the differences are of no great moment, as equally good performances can be got with some variation of area, provided the power, speed and revolutions are not high. This is because larger areas have thinner blades, the extra frictional losses being balanced by the higher efficiencies attributable to thin blades. In this connection it may be noted that the *Drake,* a highly successful vessel, with .37 D. A. R., was followed by a class of cruisers (the "county" ships) with larger values of area ratio, and equally good results.

Another point, to which attention should be drawn, is that in all cases in the table there is no restriction on diameter through limited draft or clearances. When such exists the proper area for use with the small diameter can be found from the formula in the manner illustrated by the folowing example, an actual case of a single-screw channel steamer in which the draft limited size of screw to a great extent. The vessel was of block coefficient about

* A large number in the *Journal of the American Society of Naval Engineers,* 1907 to date.

FORMULA: DISK AREA RATIO $= C + \frac{R}{10{,}000}\sqrt[4]{\frac{D\,H}{S}}$ { R=REVS. PER MIN. D=DIAMETER. H=HORSEPOWER (PER SCREW). S=SPEED; C=.30.

No.	Name.	Total I. H. P. or B. H. P.	Speed Knots.	Average Revs. Per Min.	Propeller. No. of Blades.	Propeller. Diam.	Propeller. Pitch.	Actual Disk Area Ratio.	Calculated D. A. R. by Formula.	Remarks.
1	U. S. B. S. North Dakota	31300	21.66	280	3	13.0	10.3	.54	.57	Twin screw, turbine.
2	U. S. B. S. Arkansas	28530	21.05	324	3	10.0	8.2	.58	.55	Quad screw, turbine.
3	U. S. B. S. Wyoming	31440	21.22	319	3	10.0	8.2	.58	.55	Quad. screw ,turbine.
4	U. S. B. S. Florida	40500	22.08	364	3	9.17	8.5	.62	.59	Quad. screw, turbine.
5	H. M. Battleship	about32000	21.50	310	3	9.6	8.5	.51	.54	Quad. screw, turbine (approx. figures).
6	U. S. Scout Chester	23960	25.90	603	3	6.0	6.0	.67	.67	Quad. screw, turbine.
7	Chinese Cruiser Fei-Hung	7490	20.30	535	3	5.35	5.17	.61	.57	Triple screw, turbine.
8	U. S. T. B. D. Warrington	12846	30.12	641	3	6.67	6.17	.66	.69	Twin screw, turbine.
9	U. S. T. B. D. Flusser	11270	30.41	798	3	5.25	4.83	.69	.70	Triple screw, turbine.
10	U. S. T. B. D. Cassin	15310	30.14	602	3	7.33	6.67	.68	.66	Twin screw, turbine.
11	U. S. Scout Salem	19200	25.95	378	3	9.50	8.7	.62	.59	Twin screw, turbine.
12	U. S. T. B. D. Beale	11800	29.65	783	3	5.25	4.83	.67	.70	Triple screw, turbine.
13	U. S. T. B. D. Preston	10920	29.18	779	3	5.00	4.75	.68	.69	Triple screw, turbine.
14	U. S. T. B. D. Lamson	10600	28.60	745	3	4.83	5.00	.69	.67	Triple screw, turbine.
15	U. S. T. B. D. Drayton	15520	30.80	844	3	5.25	4.83	.69	.76	Triple screw, turbine, (propulsive effic. low).
16	U. S. T. B. D. Benham	16850	29.59	585	3	7.70	6.67	.68	.70	Twin screw, turbine.
17	U. S. T. B. D. Perkins	11670	29.76	593	3	6.58	6.35	.65	.65	Twin screw, turbine.
18	U. S. T. B. D. Monaghan	12410	30.45	817	3	5.25	4.83	.68	.72	Triple screw, turbine.
19	U. S. T. B. D. McCall & Burrows	13163	30.66	816	3	5.27	4.83	.71	.73	Triple screw, turbine.
20	U. S. T. B. D. Patterson	12622	29.69	782	3	5.25	4.83	.67	.71	Triple screw ,turbine.
21	Channel Steamer	11000	22.5	460	3	6.7	5.09	.57	.57	Triple screw, turbine.
22	Passenger Str. Camden	4530	19.	552	3	5.	4.5	.51	.54	Triple screw, turbine.
23	Creole	7180	16.71	243	3	10.4	8.2	.43	.47	Twin screw, turbine.
24	Center screw of Olympic	16000	22.0	165	4	16.5	....	.57	.57 (C=.4)	Triple screw, combination (approx. figures).
25	Center screw of Liner	6700	17.6	250	3	10.0	9.5	.59	.60 (C=.4)	Triple screw , combination.
26	H. M. S. Drake	31400	24.1	123	3	19.0	23.0	.37	.43	Twin screw, reciprocating.
27	H. M. S. "County" Class	21500	23.2	141	3	15.8	19.5	.41	.43	Twin screw, reciprocating.
28	U. S. B. S. Texas	28373	21.05	124.6	3	18.65	20.0	.37	.43	Twin screw, reciprocating.
29	U. B. S. Delaware	26500	21.42	126.8	3	18.25	19.75	.40	.43	Twin screw, reciprocating.
30	U. S. Scout Birmingham	14700	24.00	191	3	12.5	15.3	.41	.45	Twin screw, reciprocating.
31	Liner	20000	21.7	90	3	19.5	28.0	.40	.39	Twin screw, reciprocating.
32	Liner	22000	19.5	90	4	21.0	24.0	.39	.39	Twin screw, reciprocating.
33	U. S. Collier Proteus	7200	14.7	102	3	16.5	16.4	.42	.38	Twin screw, reciprocating.
34	U. S. Collier Nereus	6900	14.6	99	3	16.5	16.4	.42	.38	Twin screw, reciprocating.
35	U. S. Collier Orion	6940	14.5	95	3	16.5	18.0	.38	.38	Twin screw, reciprocating.
36	U. S. Collier Mars	4150	13.2	99	4	14.75	14.25	.40	.37	Twin screw, reciprocating.
37	Motor Boat Dixie II	170	26.0	1120	3	1.95	2.67	.54	.51	Single screw, gasoline (petrol) racer.
38	U. S. Oil Barge (Gov't Ship)	178	6.23	206	3	6.75	3.5	.39	.38	Single screw.
39	Yacht	330	11.2	146	4	7.0	10.0	.35	.36	Single screw, reciprocating.
40	Yacht	101	9.1	184	4	4.63	6.25	.39	.35	Single screw, reciprocating.
41	Yacht	237	10.2	124	4	7.75	10.0	.31	.34	Single screw, reciprocating.
42	Yacht	2600	15.0	164	4	9.3	10.4	.42	.39	Twin screw, reciprocating.

.60, horsepower 3,200, revolutions 90, and desired speed 15 knots. On solving for the approximate best dimensions by the methods of the writer's paper on "Propeller Design" (International Marine Engineering, June and July, 1913), the diameter (unrestricted) comes out at 17.5 feet. By the formula the disk area ratio results—

$$D.\,A.\,R. = .30 + \frac{90}{10{,}000}\sqrt[4]{\frac{3200 \times 17.5}{15.0}} = .37$$

With 13 feet diameter (the maximum allowable) this corresponds to an area of only about 49 square feet; but on 17.5 feet 90 square feet is given by .37. This area (90 square feet) was found to be the best after several trials, on 13 feet diamter. For really good propulsive results the solution here was, of course, the substitution of twin for single-screw machinery.

The center screws on the type of ships with combined reciprocating and turbine engines have relatively great power put through them. Here the value of the constant *C*, instead of about .28 to .34, should be increased to .35 or .40 to allow sufficient area. This is because the thickness for strength requirements is then considerable, and hence breadth of blade has to be large to give a transverse sectional shape of fair efficiency.

The case of towboat propellers is obviously a special one; much larger area being needed for the requirements of maneuvering and steady gripping power with heavy loads in foul weather than would be needed for a similar vessel running "free." Excessively large areas have often been entailed by the use of cast iron wheels with necessarily very thick blades.

It may be pointed out that considerable latitude seems to exist in the possible thrusts per square inch of surface; and that by this method or by allowing for tip velocities no consistent rules of anything like equal accuracy to the above formula have been deduced.

In conclusion, the writer would draw attention to the great variety of ships dealt with in the above table. The range covers from .3 to .7 disk area ratio, and includes a barge, racing motorboat, destroyers, liners, battleships, etc.

Recent Launchings

The twin-screw lighthouse tender *Rose,* under construction for the Lighthouse Service on the coasts of Washington and Oregon by the Anderson Steamboat Company, Seattle, Wash., was launched on February 19. The hull is of steel, 127 feet 6 inches long overall, 24 feet 6 inches molded beam, 11 feet depth of hold, and at a draft of 7 feet displaces 390 tons. Propulsion will be by two sets of triple expansion reciprocating engines supplied with steam by two oil-fired watertube boilers.

The American-Hawaiian steamship *Arborean,* 404 feet long, designed for the lumber trade, was launched on February 29 at Sparrows Point, Md., by the Maryland Steel Company.

The steel hull for the derrick *Evelyn,* building for the Newton Supply Company by the Clinton Shipbuilding Company, Philadelphia, Pa., was launched on March 2.

The Wilmington Steamboat Company's new steamer, *City of Camden,* 110 feet long and of 750 gross tons and 15 knots speed, was launched by the Harlan & Hollingsworth Corporation at Wilmington, Del., on March 4.

The United States torpedo boat destroyer *Sampson* was launched at the yards of the Fore River Shipbuilding Corporation, Quincy, Mass., on March 4. The vessel was christened by Miss Marjorie Sampson Smith, daughter of Captain Roy C. Smith, U. S. N., and the eldest granddaughter of Admiral Sampson.

TWIN-SCREW PASSENGER AND MAIL STEAMSHIP OF 15,600 TONS DISPLACEMENT, BUILT AT AMSTERDAM, HOLLAND

Fig. 1.—S. S. *Jan Pieterszoon Coen*

New Dutch Passenger and Mail Steamship

Twin-Screw Vessel Built by the Nederlandsche Scheepsbouw Maatschaapij for the Netherland Steamship Company

BY FREDERICK C. COLEMAN

THE largest vessel built at any Dutch shipbuilding establishment is the twin screw passenger and mail steamer *Jan Pieterszoon Coen,* recently constructed by the Nederlandsche Scheepsbouw Maatschappij for the Netherland Steamship Company, and illustrated in the engravings on this and the following pages. This vessel is now in service between Amsterdam and the Dutch East Indies. Its dimensions are as follows:

Length, overall .....522 feet 6 inches
Length between perpendiculars.....503 feet 6 inches
Breadth, molded .....60 feet 4 inches
Depth, molded to shelter deck.....39 feet
Draft, loaded .....25 feet
Displacement .....15,600 tons
Deadweight capacity .....6,950 tons

There are seven decks, three being continuous over the whole length. Eight watertight bulkheads reach to the upper deck, with the exception of the collision bulkhead, which reaches to the shelter deck, and they divide the ship into nine watertight compartments. The ten watertight doors can be closed in case of emergency from the bridge by means of Stone-Lloyd's patent pneumatic hydraulic closing apparatus.

The propelling machinery consists of two triple expansion engines with cylinders 27, 44 and 75 inches in diameter, and 49¼ inches stroke, developing 6,000 indicated horsepower at 85 revolutions per minute. The speed is 15 knots at 25 feet draft. Steam is delivered at 210 pounds per square inch gage pressure by eight single-ended Scotch boilers having two furnaces each and a total grate surface of 520 square feet, and a heating surface of 21,000 square feet. The boilers have Howden's forced draft. The engines and boilers were manufactured by the Nederlandsche Fabrik van Werktuigen and Spoorweg-Materieel, of Amsterdam.

The steering mechanism consists of two engines, each of which may be used separately. One of the engines is a Hele-Shaw-Martineau patent electric hydraulic steering gear, and the other is a steam steering gear of Hastie's pattern. Both of these engines may be worked from the bridge by telemotor control.

The ship has two masts, the foremast carrying four

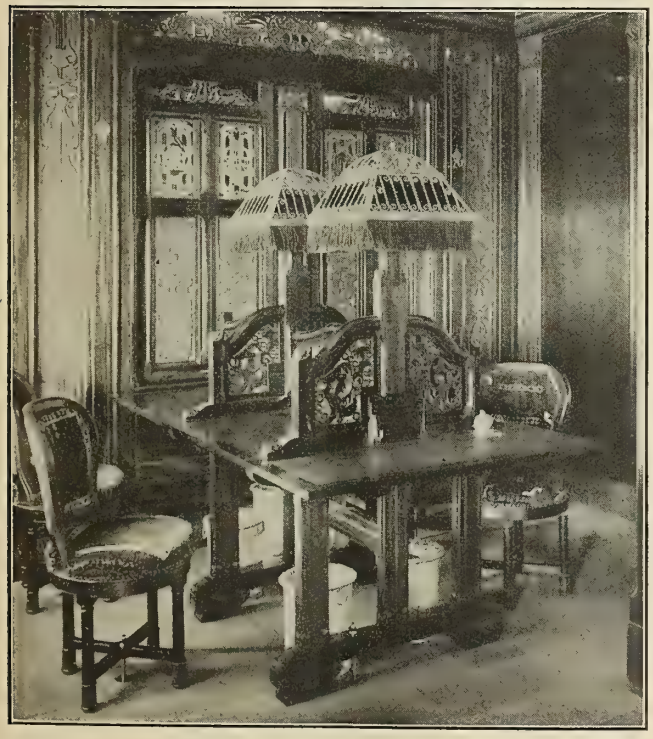


Fig. 2.—First-Class Writing Room

Fig. 3.—En Suite Staterooms

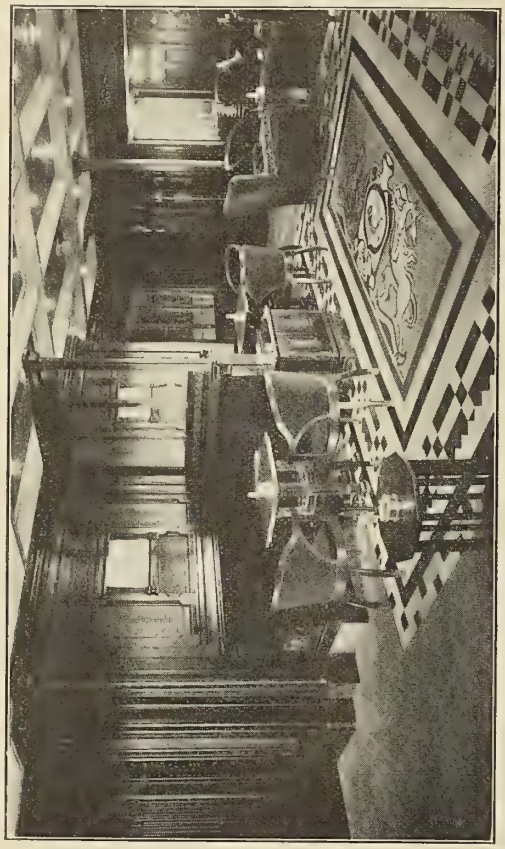


Fig. 5.—First-Class Smoking Room

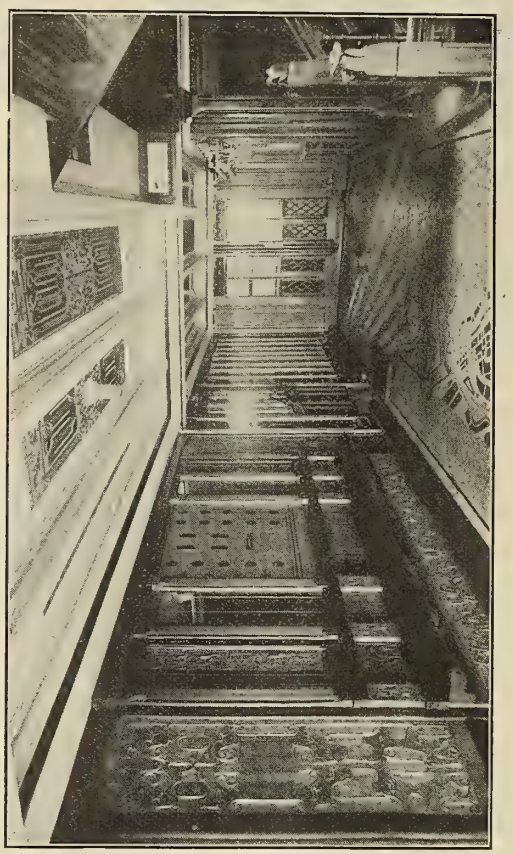


Fig. 7.—First-Class Lobby

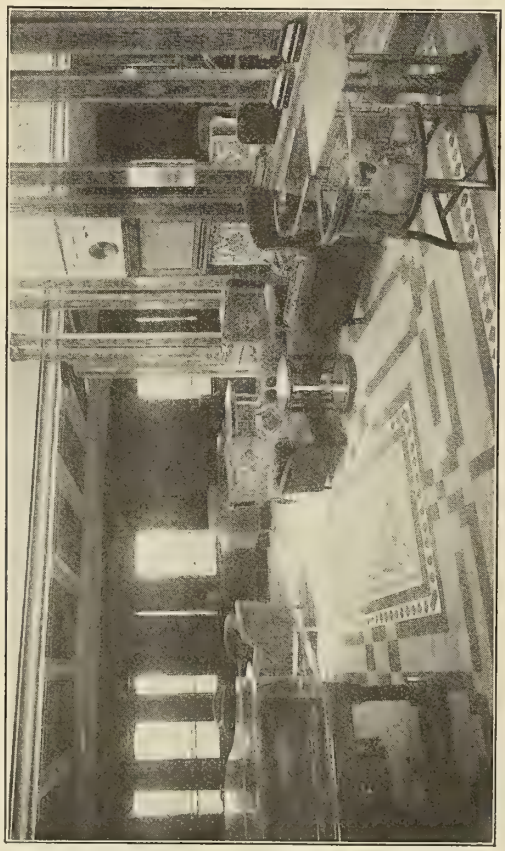


Fig. 4.—First-Class Saloon

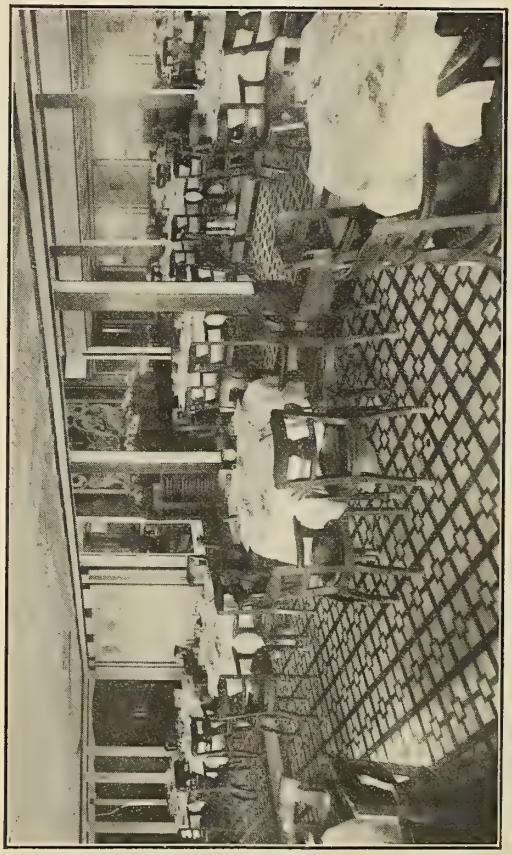


Fig. 6.—First-Class Dining Room

derricks of 6 tons lifting capacity each, and there is also a 25-ton derrick provided here to take specially heavy loads. The two derrick posts on the bridge deck have two 3-ton derricks. The derricks are worked by six steam winches, and in addition there are eight hydraulic cranes.

There is accommodation for 202 first class passengers in 107 cabins and four suite rooms, 129 second class passengers in 49 cabins, 46 third class passengers in 16 cabins and 42 fourth class passengers, and a crew of 261, making accommodation for a total number of persons on board of 680. As will be gathered by reference to the illustrations, the passenger apartments are most luxuriously equipped. In addition to the four suite cabins, each of which has a separate bath room, there are several saloons at the disposal of the first class passengers. These include a deck saloon 10 feet high, a smoking room 10 feet high, dining saloon with 138 seats, a verandah café, a nursery, gymnasium, and photographers' dark room.

The vessel is fitted with a Marconi wireless installation and submarine bells, and it is electrically lighted throughout. There are fourteen boats, inclusive of two motor boats, and all of these may be lowered by means of Welin quadrant davits. They provide sufficient room for all persons on board.

Merchant Shipbuilding in France

Since the outbreak of the European war the building of merchant vessels in the private shipyards of France has been greatly diminished, owing to the application of every available resource to military service. At the outset about 11 percent of the population of France was mobilized, leaving for commercial work only men either under eighteen or over fifty years of age and those incapacitated for military service. Furthermore, the invasion of northern France, which furnishes 80 percent of the nation's products from the mines and metal industries, has made the shipbuilders in France practically dependent upon England for shipbuilding materials.

In spite of these obstacles, however, the French shipbuilders are to be complimented upon the progress they have made, as shown by the following statement covering work finished or in hand since August 2, 1914:

At Dunkerque, the Ateliers & Chantiers de France has completed the passenger boat *Asie*.

At Havre, the Forges & Chantiers de la Mediterranée has completed the cargo boats *Conde* and *Belle Isle* and the passenger boat *Versailles*.

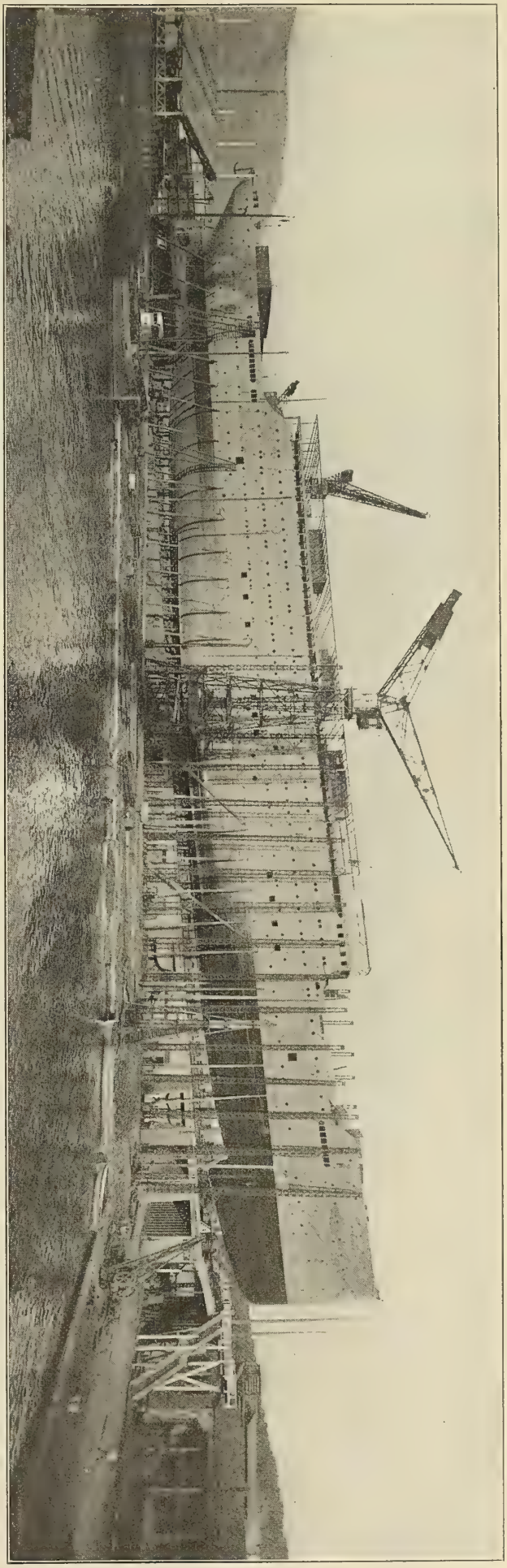
At Rouen, the Ste Ayme des Chantiers de St. Nazaire has completed the oil tanker *Meuse* of 7,400 tons deadweight and the cargo vessels *Ohio* and *Omnium* of 12,200 tons deadweight.

At St. Nazaire the above firm has under construction the passenger steamer *Paris,* of 36,650 tons, and the Chantiers de la Loire has built the passenger steamer *Sphinx*.

At Nantes, the Ateliers & Chantiers de la Loire has built the cargo boats *Chateau Lafitte, Saint Jean* and *Chateau Palmer,* and has in hand the *Chateau Latour* and another cargo boat of 4,000 tons.

At La Ciotat, the Chantiers & Ateliers de la Provence has completed the passenger steamer *Lafayette* and has under construction two passenger steamers of 780 tons and a cargo boat of 5,000 tons.

At La Seyne, the Forges & Chantiers de la Mediterranée has built the cargo boats *Gharb* and *Aurogny* and the passenger steamers *Providence* and *Massilia*.



S. S. *Providence* under Construction at the Yards of the Forges & Chantiers de la Mediterranée at La Seyne, France

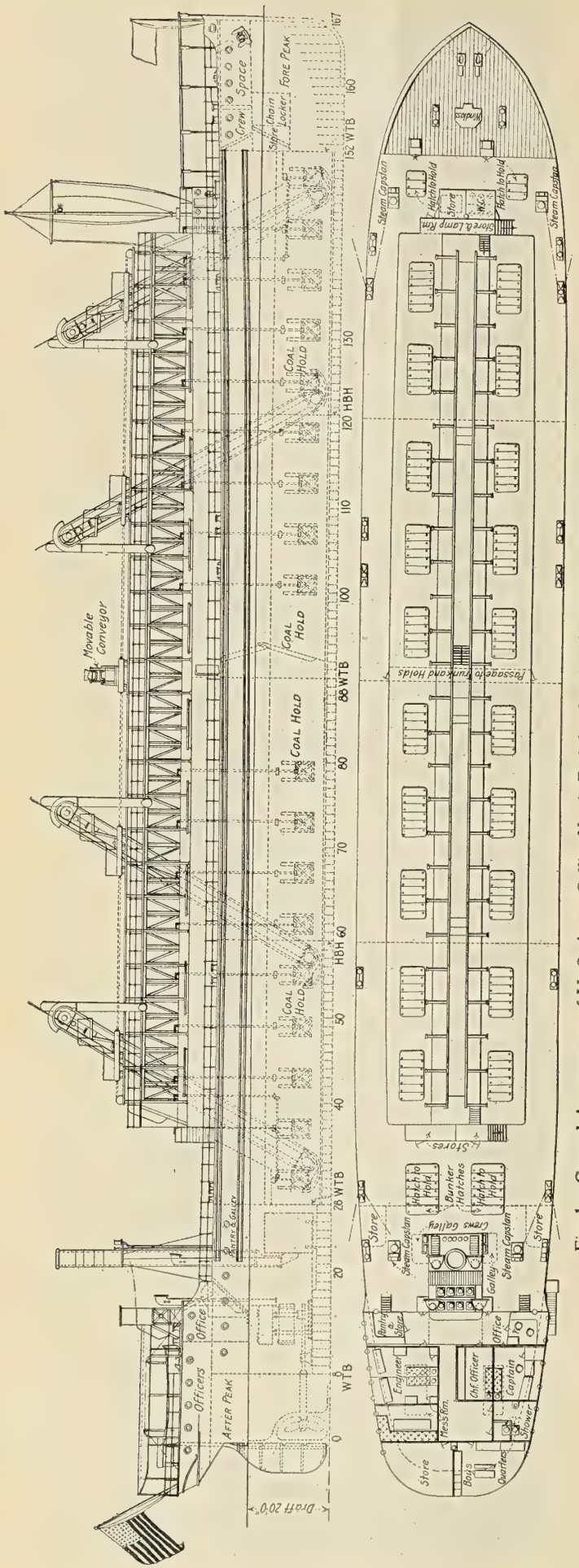


Fig. 1.—General Arrangement of U. S. Army *Collier No. 1*, Built by the Shanghai Dock & Engineering Company, Ltd.

United States Army Collier Built in China

Over a year ago there was completed at the yards of the Shanghai Dock & Engineering Company, Ltd., Shanghai, China, for the quartermaster's department of the United States army, Philippines division, Manila, P. I., a single screw steel collier of about 6,000 tons deadweight capacity on a draft of 20 feet. The vessel was built to meet the general requirements of a harbor coal hulk and was fitted with four coal bunkering elevators, coal shoots and weighing machines and one elevating coal conveyor capable of discharging at a height of 50 feet above the deep load waterline of the collier. The auxiliary propelling machinery was located aft, together with the pumps and turbine driven electric generating sets.

The principal dimensions of the vessel are as follows:

Length overall	362 feet
Length between perpendiculars	350 feet
Beam, molded	50 feet
Depth, molded	28 feet
Capacity of holds, including bunkers	6,000 tons

The hull has a straight stem and elliptical stern and

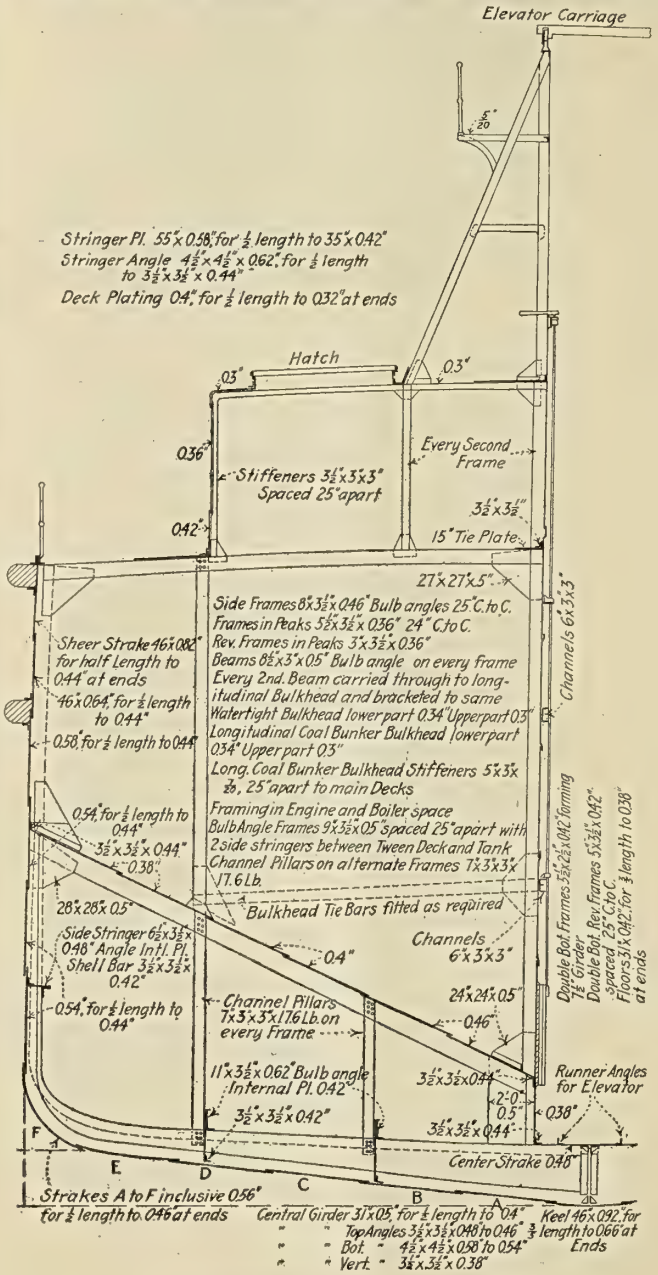

Elevator Carriage
Stringer Pl. 55"x0.58" for ½ length to 35"x0.42"
Stringer Angle 4½"x4½"x0.62" for ½ length to 3½"x3½"x0.44"
Deck Plating 04" for ½ length to 032" at ends
Hatch
03"
03"
036"
Stiffeners 3½"x3"x3" Spaced 25" apart
Every Second Frame
3½"x3½"
042"
15" Tie Plate
27"x27"x5"
Side Frames 8"x3½"x046" Bulb angles 25" C. to C.
Frames in Peaks 5½"x3½"x036" 24" C. to C.
Rev. Frames in Peaks 3"x3½"x036"
Beams 8½"x3"x05" Bulb angle on every frame
Every 2nd. Beam carried through to longitudinal Bulkhead and bracketed to same
Watertight Bulkhead lower part 034" Upper part 03"
Longitudinal Coal Bunker Bulkhead lower part 034" Upper part 03"
Long. Coal Bunker Bulkhead Stiffeners 5"x3"x 7/16, 25" apart to main Decks
Framing in Engine and Boiler space Bulb Angle Frames 9"x3½"x05" spaced 25" apart with 2 side stringers between Tween Deck and Tank
Channel Pillars on alternate Frames 7"x3"x3" 17.6 Lb.
Sheer Strake 46"x082" for half Length to 044" at ends
46"x064" for ½ length to 044"
058" for ½ length to 044"
054" for ½ length to 044"
3½"x3½"x044"
038"
28"x28"x05"
Bulkhead Tie Bars fitted as required
Side Stringer 6½"x3½"x 046" Angle Intl. Pl. Shell Bar 3½"x3½"x 042"
Channels 6"x3"x3"
04"
054" for ½ length to 044"
Channel Pillars 7"x3"x3"x17.6 Lb. on every Frame
24"x24"x05"
046"
11"x3½"x062" Bulb angle Internal Pl. 042"
3½"x3½"x044"
Runner Angles for Elevator
2'-0"
05"
038"
3½"x3½"x042"
3½"x3½"x044"
Center Strake 048"
F
E
D
C
B
A
Strakes A to F inclusive 056" for ½ length to 046" at ends
Central Girder 31"x05" for ½ length to 04"
" " Top Angles 3½"x3½"x048" to 046"
" " Bot. " 4½"x4½"x058" to 054"
" " Vert. " 3½"x3½"x038"
Keel 46"x092" for ½ length to 066" at Ends
Channels 6"x3"x3"
Double Bot. Frames 3½"x3½"x042" forming T.E. Girder
Double Bot. Rev. Frames 5"x3½"x042" spaced 25" C. to C.
Floors 31"x042" for ½ length to 038" at ends

Fig 2.—Section in Way of Hold

Fig. 3.—Launch of U. S. Army *Collier No. 1* at Chinese Shipyard

there are four transverse watertight bulkheads, all extending to the main deck, as well as longitudinal coal division bulkheads for the elevator well and partial coal hold bulkheads as shown on the general arrangement plans and midship section. A double bottom extends from frame 28 to frame 152. The center girder is continuous 31 inches by .5 inch for one-half length amidships to 37 inches by .4 inch at the ends, with side well girders as shown on the plans. The flat keel is 46 inches wide by .92 inch thick for three-fifths of the vessel's length amidships tapered to .66 inch at the ends. There are two longitudinal girders on each side of the centerline formed of plate .42 inch thick with bulb angles 11 inches by 3½ inches by .62 inch on top of the floors.

The frames are 5½ inches by 3½ inches by .42 inch and the reverse frames 5 inches by 3½ inches by .42 inch, forming 7½-inch girders spaced 25 inches apart. The shell plating from the flat keel to the upper turn of the bilge is .56 inch thick for half the vessel's length amidships to .46 inch at the ends. From the upper turn of the bilge to the strake in way of the tank top at the side the plating is .54 inch thick amidships reduced to .44 inch at the ends. The sheerstrake is 46 inches by .82 inch for one-half length to .44 inch at the ends, while the strake below is .63 inch thick reduced to .44 inch at the ends.

The stem is an iron forging 10 inches by 2⅝ inches built according to Lloyd's rules. The stern post is 9 inches by 7 inches, reduced from the lower part of the rudder trunk to its head, where it is 7½ inches by 7 inches, secured to the transom floor plate by two vertical angles. The propeller post is 10 inches by 7 inches, bossed to take the propeller shaft. The rudder is of the single plate type with cast steel stock.

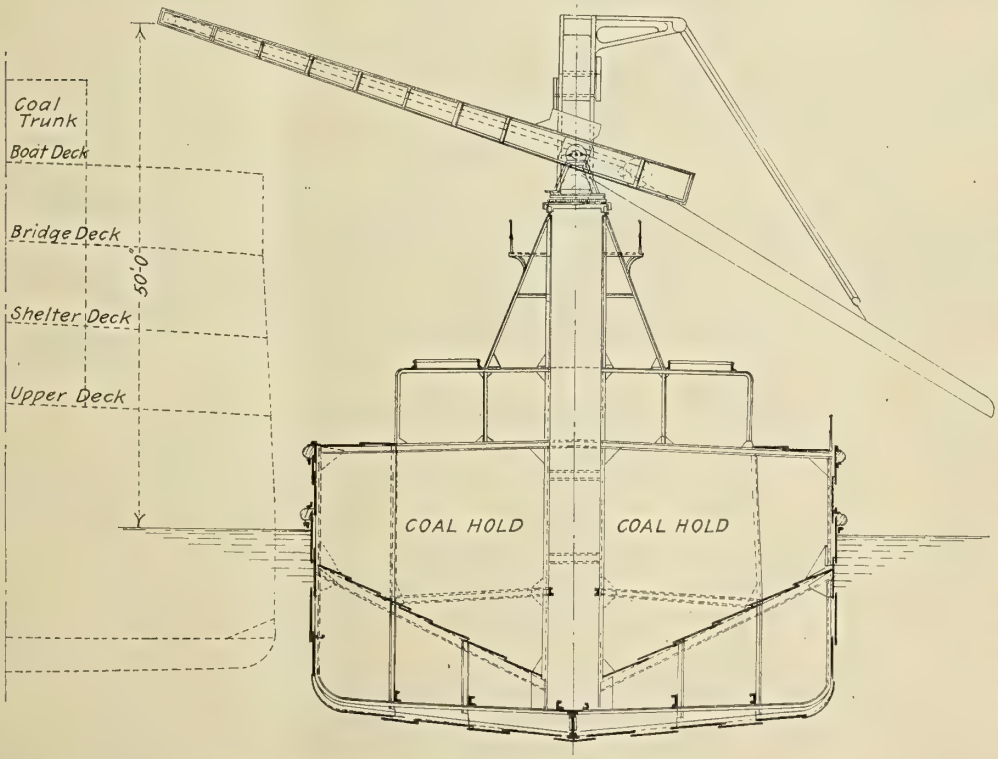


Fig. 4.—Midship Section, Showing Coaling Apparatus

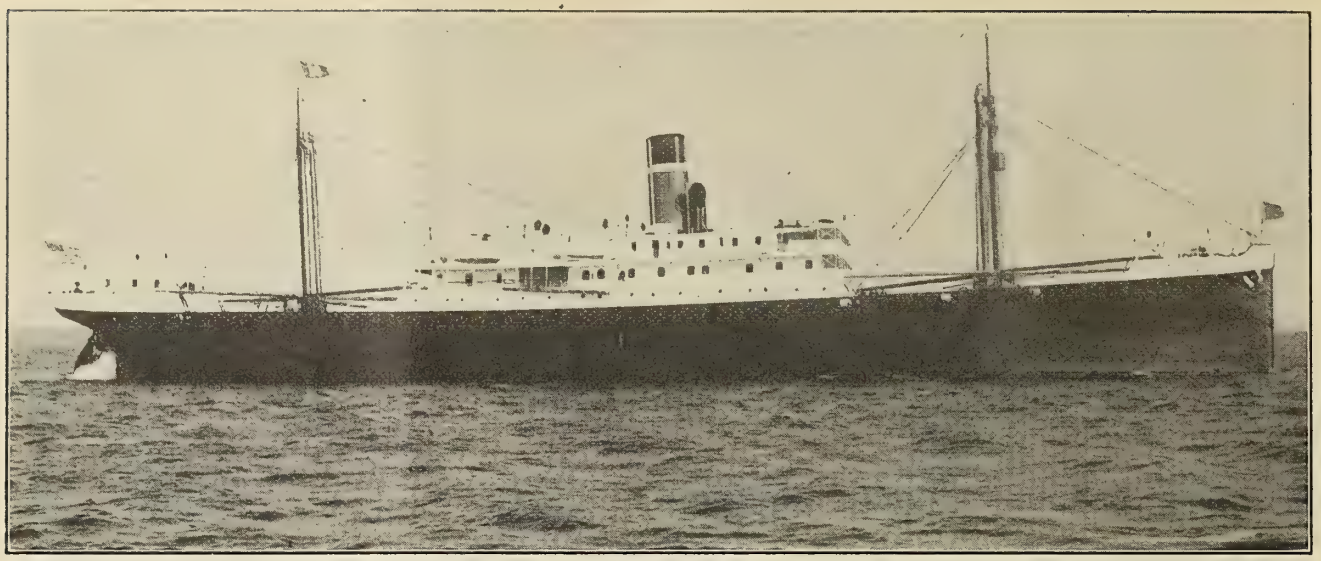


Fig. 1.—Munson Steamship *Munamar*, Built by the Maryland Steel Company, Sparrows Point, Md.

Designed for the passenger and cargo traffic between New York and the island of Cuba, the *Munamar* has been built with special regard to the comfort of passengers traveling in tropical climates, as well as for a rapid and economical cargo carrier. With a deadweight carrying capacity of 4,000 tons, she has averaged a speed of 14½ knots from dock to dock on the voyage between New York and Antilla, Cuba. During the trip from Nuevitas to New York, ending February 9, the vessel covered the distance of 1,280 miles in the record time of three days and nine hours.

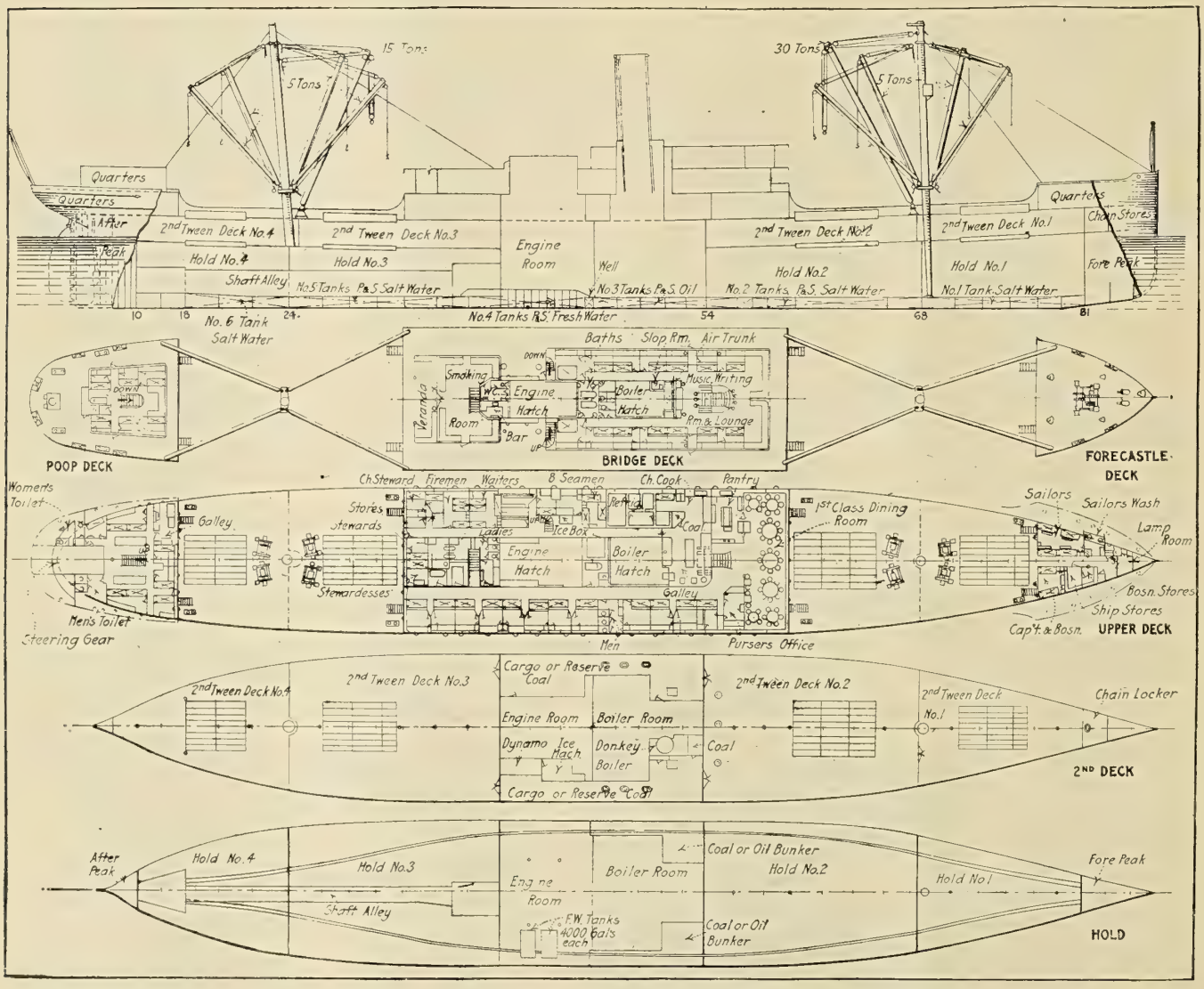


Fig. 2.—General Arrangement of the *Munamar*, Showing Profile and Deck Plans

Munson Steamship Munamar

The new Munson liner *Munamar,* recently built by the Maryland Steel Company, Sparrows Point, Md., is a steel vessel of 4,000 tons deadweight capacity built on the Isherwood system of longitudinal framing with a raised poop, forecastle and saloon deck amidships. She was designed by Theodore E. Ferris, naval architect, New York, in collaboration with the builders, and has accommodations for fifty-two first class and thirty-eight second class passengers besides a crew of sixty-four and ten officers. Her designed speed at sea is fifteen knots loaded.

The principal dimensions of the *Munamar* are as follows:

Length overall	368 feet 3 inches
Length between perpendiculars	353 feet 9 inches
Beam, molded	47 feet 3 inches
Depth, molded	27 feet 3 inches
Load draft	21 feet 1 inch

The general arrangement of the accommodations is shown on the accompanying plans. At the forward end of the midship house on the main deck is the first class dining saloon, which seats forty-eight persons. This saloon is finished in white enamel with tapestry panels, while the floor is covered with asbestolith. Aft of the dining saloon on the starboard side are twelve first class staterooms, while on the port side of the main deck amidships are quarters for the engineers. At the stern are the second class accommodations. Above on the bridge deck are fourteen first class staterooms and the first class social hall paneled in sycamore. The first class smoking room, located on the bridge deck, is paneled in quartered oak, while the floor is laid with rubber tile. Hot and cold fresh and salt water is supplied to all of the baths and toilets.

For handling cargo both the main mast and the fore mast are fitted with four five-ton and one thirty-ton cargo booms, the booms on each mast being operated by four reversible double-cylinder single drum winches.

Propulsion is by a triple expansion inverted reciprocating engine with cylinders 27¼, 46 and 78 inches diameter by 51 inches stroke, supplied with steam at 200 pounds per square inch pressure by three Scotch boilers 16 feet 6 inches diameter by 11 feet 6 inches long, equipped with White mechanical fuel oil burners. The boilers are operated on the Howden system of forced draft, the forced draft blowers being supplied by the American Blower Company.

The engine room auxiliaries include a complete outfit of Blake and Knowles pumps, two 15-kilowatt electric generating sets driven by Terry turbines, a two-ton refrigerating plant supplied by the Brunswick Refrigerating Company, a Shutte & Koerting evaporator and a Shutte & Koerting distiller of 4,000 gallons capacity. The feed water heater is of a Shutte & Koerting type, while hot water for baths is supplied by a Goubert heater made by the Griscom-Russell Company. Ventilation of the hull is by Sturtevant fans.

In service the *Munamar* has proved remarkably steady, with no perceptible vibration. From New York to Antila, Cuba, she has averaged 14½ knots on reduced revolutions from dock to dock.

Pressure on a Vessel's Bottom During Launching

BY G. H. BARBER

Before launching a vessel there are two pressures, in addition to the locally concentrated pivoting pressure, which should be carefully determined both as to location and amount. These are the pressure on the ground ways and the thrust of the packing against the shell. To ensure a successful launching, an excess of these pressures beyond a safe amount must be guarded against by the stiffening of the ground way supports and the fitting of special interior shoring in the vessel.

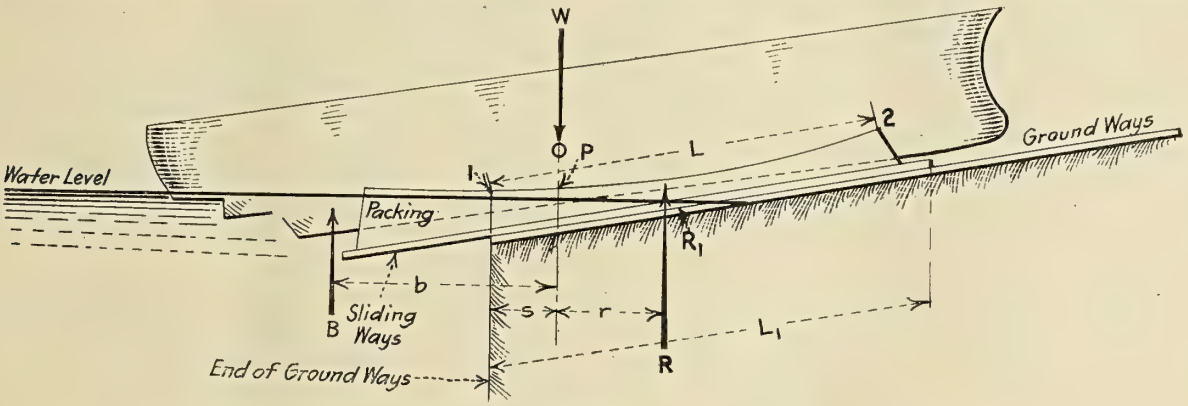


Fig. 1

The amounts of these pressures are easily obtained until the weight of the vessel commences to be taken up by the buoyancy, and until the after end of the packing passes the end of the ground ways. The pressure on the ground ways at the start is, of course, the total weight on the ways, divided by the effective area of the sliding ways. The thrust against the shell at the start may be taken, for all practical purposes, as equal to the weight of the vessel divided by the area of packing in contact with the shell.

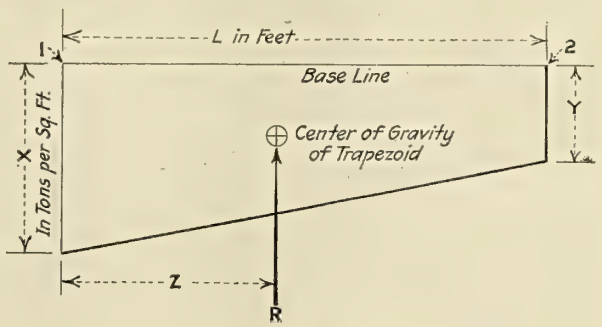


Fig. 2

It is with the maximum pressures that we are chiefly concerned. A method which gives very close results for the thrust exerted at any point of the shell in contact with

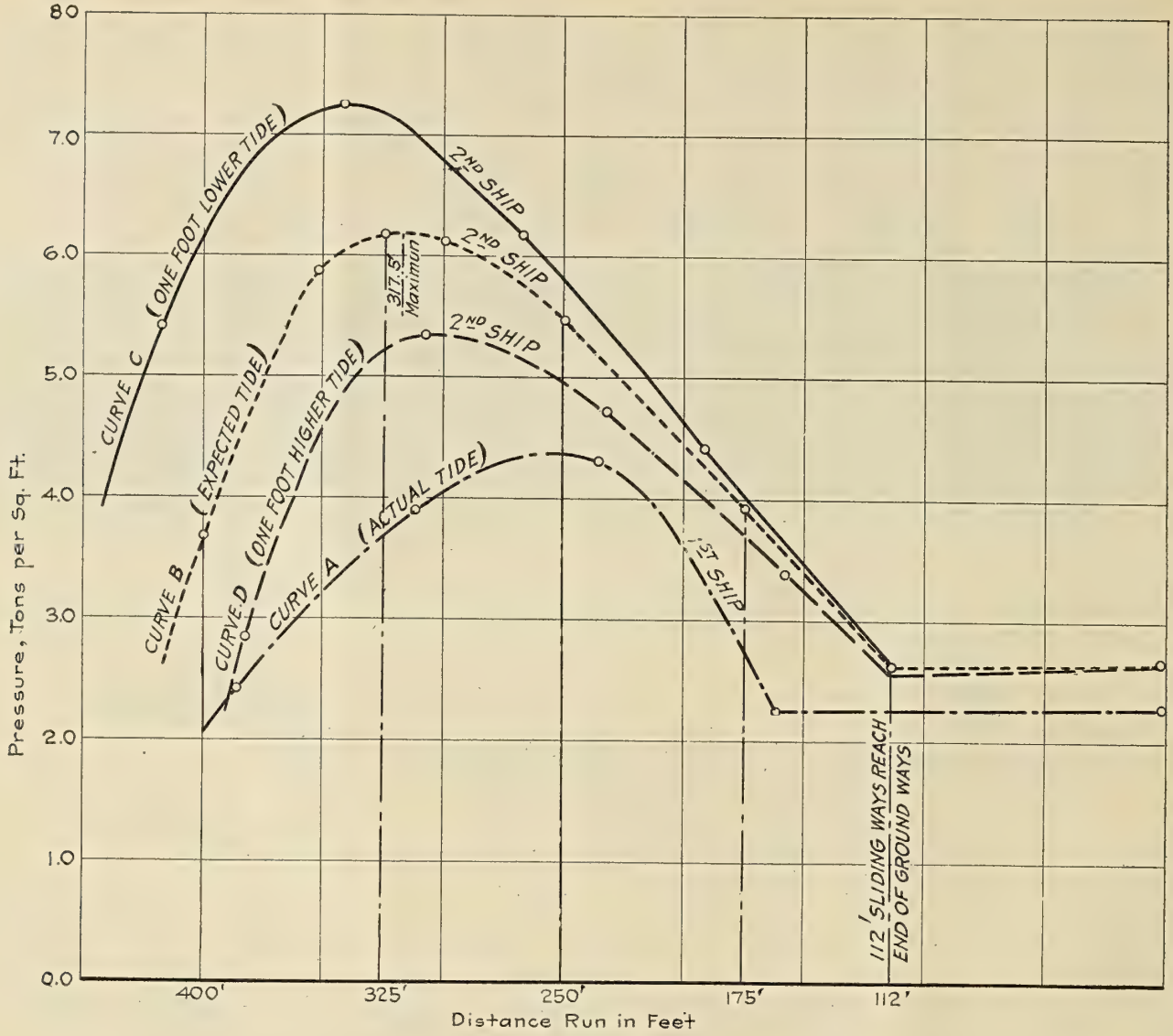


Fig. 3

the packing is given below. The pressure on the ground ways is similarly obtained, but with the additional consideration of the weight of the packing and the slight difference in the bearing area.

Fig. 1 shows the forces at work and the data required. The values for B and b are usually available from the preliminary launching calculations. If not, they can be readily worked out, and curves of buoyancy and fore-and-aft center of buoyancy plotted (see article on "Launching Calculations," in December, 1914, International Marine Engineering). The launching weight and longitudinal center of gravity of the vessel must also be calculated. The values for S and L are dependent on the run of the vessel relative to the end of the ground ways.

Referring to Fig. 1, it is evident that the pressure is greater at 1 than at 2, due to the weight of that part of the vessel overhanging the end of the ground ways, which weight is only partially supported by buoyancy. If the vessel is considered to be rigid, the pressure curve will be a straight line between the points 1 and 2. The area enclosed between this curve and its base line will be trapezoidal in shape and equal to R. R will act at the center of gravity of the trapezoid a distance Z, Fig. 2, from the left side. Z is a fixed distance equal to $s + r$, and is found as follows:

$W - B = R$. Then taking moments about the point P,

$\Sigma M = O = B \times b - R \times r$, or

$B \times b = R \times r$, and

$$r = \frac{B \times b}{R},$$

$$Z = r + s.$$

Regarding the value and direction of R, the following should be noted: The amount of weight which the packing must support equals $W - B$. This weight is supported by the normal reaction and friction of the packing against the shell. R, therefore, being the summation of the normal reactions, does not act exactly in a vertical direction, but very nearly so. For all practical purposes, R may be taken as equal to $W - B$, acting over a length L, as shown. On the other hand, when considering the pressure on the ground ways, the ground way reaction must be taken as $(W - B) \cos \theta$—which equals R_1—acting over a length equal to the effective length of sliding ways in contact with the ground ways L_1, the angle being the angle of the ground ways to the horizontal.

There is only one trapezoid which will fill the requirements of R, Z and L for a given run. Therefore, X and Y, Fig. 2, must have definite fixed values for any definitely chosen run of the ship. These values can be obtained algebraically from the two equations, which give the area

of the trapezoid, and the center of gravity of the trapezoid from its greater side, namely:

$$(1)\quad R = L\left(\frac{X+Y}{2}\right) \text{ and } (2)\quad Z = \frac{L}{3}\left(\frac{X+2Y}{X+Y}\right)$$

The values of all terms in equations (1) and (2) are now shown except those of X and Y, which are the bottom pressures at the points 1 and 2, respectively. The values of X are obtained for different runs of the vessel, and a curve plotted on distance run in feet as a base and tons per square foot as a vertical scale. To get this latter value divide the value of X as obtained above by the total width in feet of the sliding ways.

In Fig. 3 are shown curves for two battleships of different classes which were launched from the same building slip. Curve A has been plotted for actual conditions of tide and weight for the earlier ship. Curves B, C and D are for the later and larger vessel, and show the bottom pressures, which would exist for the expected height of tide and for tides one foot lower and one foot higher than the expected tide. The actual tide for the second ship was, by a rare coincidence, very nearly that which was used in the preliminary calculations, based on the expected height of tide.

The curves A and B will bring out the difference of pressures which exist between two ships due to differences in weight, length and initial position on the ways, since the height of tide for curve A was within an inch of the tide for curve D. Curves B, C and D illustrate the effect of tide on bottom pressures.

The values for curves C and D may be quickly obtained for the values of buoyancy used for curve B by substituting in the equations (1) and (2) the values of L and Z corrected for the decreased and increased distance run, respectively, to obtain this buoyancy (see previous article mentioned above).

The maximum bottom-pressure curve is of great practical value in that it shows exactly over what part of the vessel the maximum thrust is exerted. Thus is indicated the portion of the ship which must be especially stiffened in wake of the packing if the bottom pressures exceed a safe limit.

As an example, assume that it is desired to fit special shoring to take all thrust in excess of 5 tons per square foot. Referring to curve B, it is noted that the maximum pressure is exerted after a run of 317.5 feet. At the start the center of gravity of the vessel, located at frame No. 75, the frames being spaced 4 feet apart, was 352.5 feet from the end of the ground ways. Then, since $\frac{352.5 - 317.5}{4} = 9$ frames, about, the frame No. 75 was 9 frames forward of the end of ways when the maximum pressure was exerted; that is, frame No. 84 was over the end of the ground ways. A pressure of over 5 tons per square foot is exerted from the 225-foot run to the 375-foot run, or for 150 feet. The vessel must, therefore, be shored from 317.5 feet — 225 feet = 92.5 feet aft of frame No. 84 to 375 feet — 317.5 feet = 57.5 feet forward of frame No. 84, or from about frame No. 70 to about frame No. 107.

Trials of Battleship Pennsylvania.—During her standardization trials off Rockland, Me., on February 24, the new United States battleship *Pennsylvania*, built by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., attained a speed of 21.75 knots as against her contract speed of 21 knots. The average of the five runs was 21.38 knots, and on her twelve-hour run off the Virginia capes she averaged 21.05 knots in a 50-mile gale.

Conversion of the Army Transport McClellan into a Refrigerator Ship

BY F. J. FRENCH

When, in the early summer of 1914, the exigencies of the service demanded the carriage of large quantities of frozen beef and other chilled supplies from the United States to ports on the Gulf of Mexico, the quartermaster corps of the army was directed to make such alterations to one of its transports as were necessary to fit it for this purpose.

The vessel chosen for conversion was the U. S. A. transport *McClellan*, which is 321 feet waterline length by 37 feet beam and 21 feet load draft, with a maximum displacement of about 5,000 tons. The task of converting this ship in a minimum of time and at a minimum cost was allotted to the quartermaster's office at New Orleans, with Captain W. O. Smith, U. S. A., in charge of the work. The record time which was made in putting the ship into service and her satisfactory performance since reflect credit upon the department.

This ship was of English build and had formerly been fitted as a refrigerator ship, though all the evidences had been removed with the exception of a 'thwartship refrigerator room on the lower deck abaft the engine casing and a large ice machine of about twenty tons' capacity, which it was decided was beyond repairs.

In tearing out the insulation of the old ice room the steel lower deck under the entire space from engine casing to No. 3 hatch was found to be entirely rusted out. This deck was removed and new plating laid and riveted up in thirty-six hours.

The portion of the ship available for refrigerating and cooling space consisted of the 'tween decks, divided into four parts by transverse bulkheads, and four holds, the dividing bulkheads being the same as for the 'tween-deck spaces.

These spaces were known (beginning forward) as 'tween decks spaces, Nos. 1, 2, 3 and 4, and holds Nos. 1, 2, 3 and 4, and will be referred to by these numbers. The hold spaces were broken at intervals by strong beams at the orlop deck level, and a stringer plate about 30 inches wide covered the beams at their extremities. In hold No. 4 a temporary wooden deck had been laid on these beams, and this had to be removed, as well as all living quarters in the 'tween decks.

The insulating material selected was "Waterproof Lith" supplemented by "Flax Fibre" for filling the interstices between beams and frames and for packing in other places where the Lith could not be easily fitted. It was decided to utilize the 'tween spaces for carrying chilled material only and to reserve the holds for the frozen beef, fitting the latter with a greater ratio of cooling-pipe surface.

As the meat was to be received frozen hard and covered with sacking, it was considered unnecessary to fit hanger rails and hangers, and the method of storage adopted was simply to pile the cargo in layers on a floor grating of wood, with wood gratings at suitable intervals between the layers of meat. This arrangement had the advantage not only of affording greater storage space, but also of saving a very considerable sum of money, as with the alternative method it would have been necessary to fit many thousands of feet of hanger rails and also to build in a deck at the orlop deck level of sufficient strength to support its quota of beef. As the allotment of funds for this work was limited, the saving mentioned was a consideration not to be ignored.

The construction of the insulation is shown in the cross-section (Fig. 2), and is as follows:

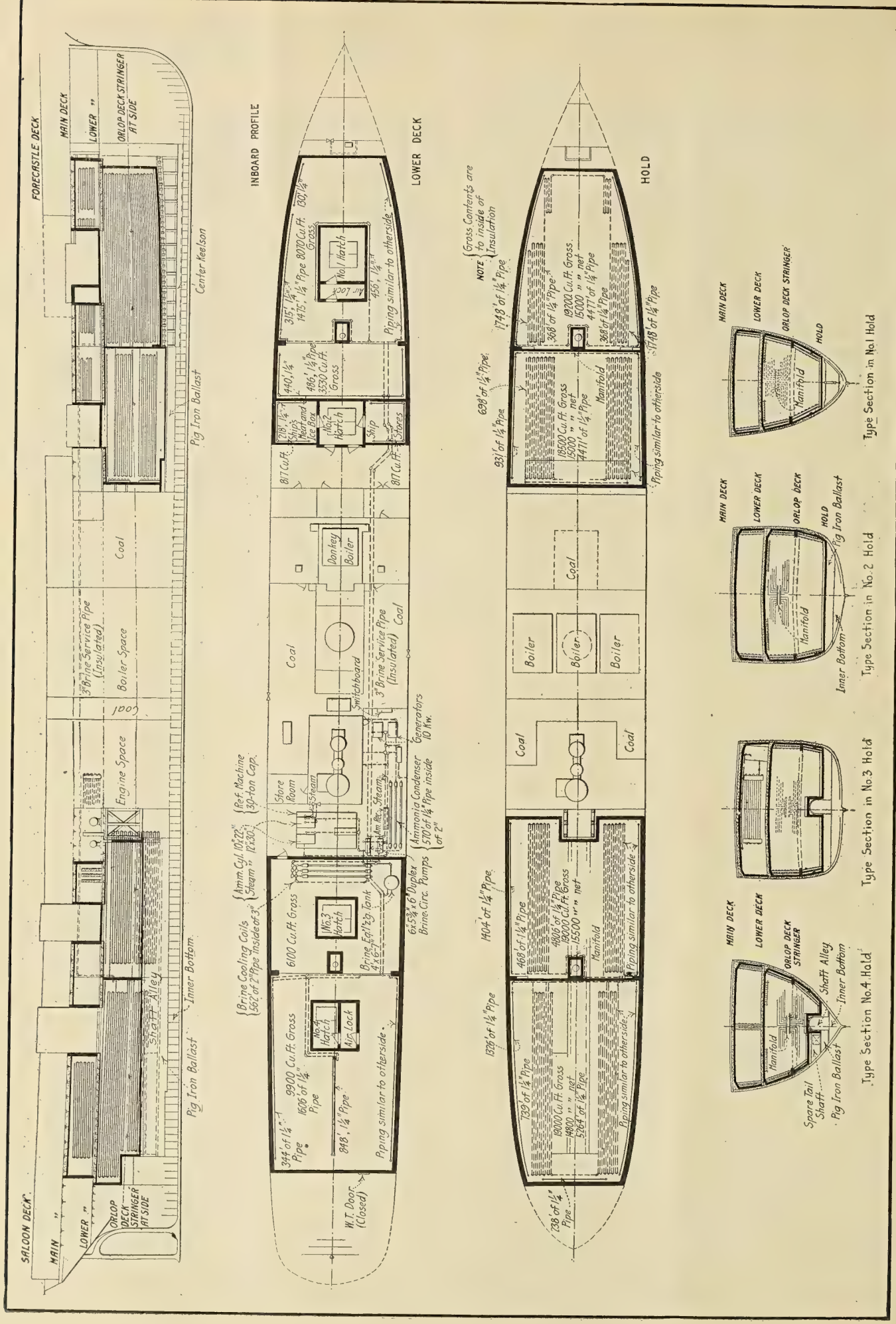


Fig. 1.—Profile, Deck Plans and Cross-Sections of the United States Army Transport, Showing Location of Refrigerator Holds and 'Tween Decks and Arrangement of Refrigerating Machinery

A 2-inch by 4-inch yellow pine stud was bolted (in suitable lengths) at head and heel to each frame. Nailed to studs was one thickness of tongued and grooved cypress sheathing. Then two layers of 90-pound roofing paper. Over the paper were placed vertical nailing strips 2 inches by 2 inches, on 20-inch centers. Between the nailing strips came the blocks of 2-inch waterproof Lith set in cement, then two more layers of paper, and finally another thickness of cypress sheathing; the inside surface of the compartment being then treated with three coats of best orange shellac.

This construction applies to the undersides of decks as

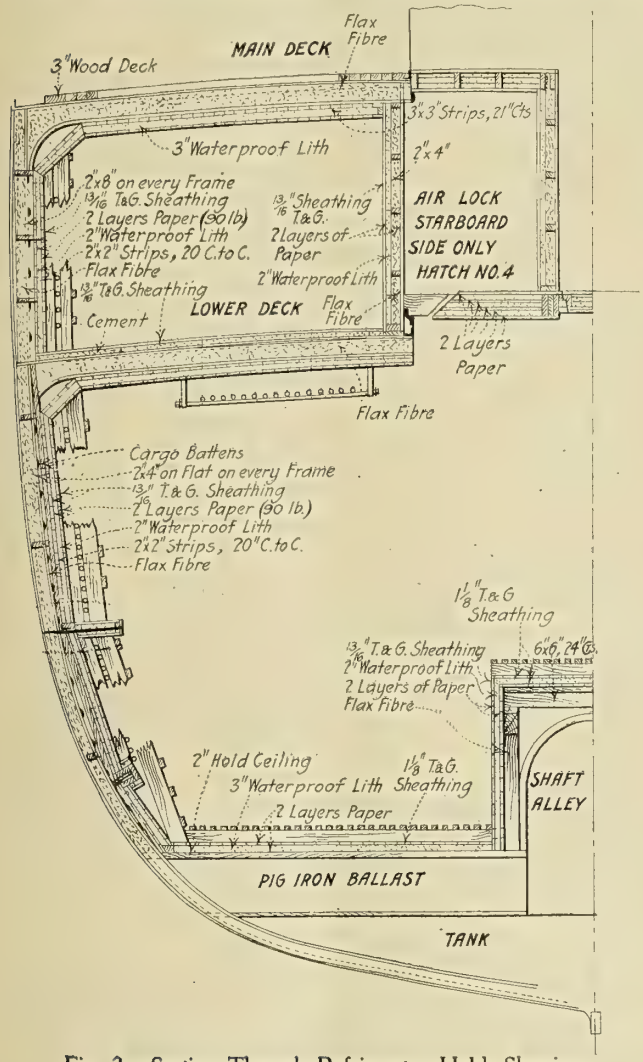


Fig. 2.—Section Through Refrigerator Hold, Showing Method of Insulation

well as to the sides of the ship. Between the wall thus formed and the decks and skin of the ship was packed Flax Fibre well tamped and supplied with spacers to prevent settling.

Under the main deck, which was to be subjected to the direct rays of a tropical sun, the thickness of the Lith was increased from 2 inches to 3 inches, and the same thickness was laid on the wood floor covering the pig iron ballast in the holds. Raised platforms or false decks were also constructed in the holds where conditions of form showed it to be expedient, and a casing was built around the shaft alley, as shown.

All hatchways were cased in and vertical and horizontal doors fitted, with air locks from the former into the 'tween-deck compartments.

The walls and ceiling of all compartments were then covered with 1¼-inch brine pipes, the total length of pipe being proportioned to the cubical contents of the compartment as given in the following table. These pipes were subdivided into groups or coils of from 500 to 600 feet in length, one end of each coil leading to the supply manifold, and the other to the suction manifold, the manifolds being placed inside the compartment on the transverse bulkhead, as near the entrance as practicable. In addition to control by the manifold valves, each compartment can be cut out from the main line. The entire system was tested out under air pressure before being filled with brine.

The brine pipes in the holds were protected from contact with the meat by aprons formed of 2-inch by 4-inch strips nailed horizontally to 4-inch by 6-inch vertical studs. The transverse steel bulkheads separating the holds were fully insulated on one side only and to lessen the heat leakage, from the skin of the ship inward into the other compartment the bare side was insulated for a width of 3 feet around the margin. The same treatment was accorded the orlop deck stringer plates, while on the bulkheads separating the 'tween-deck compartments this was the only insulation fitted, as it was not thought that it would be necessary to operate them independently of each other.

The gross volumes shown in the table are to the inside surface of the ceiling, while the net volumes are to the inside of the brine pipe aprons. The ratios of cubic feet of space per lineal foot of 1¼-inch brine pipe are based on the gross volume of the compartment, and the supply mains have been reduced to equivalent lineal feet of 1¼-inch pipe.

Compartment	Cubic Feet, Gross	Cubic Feet, Net	Lineal Feet, 1¼-In. Pipe	Ratio Cubic Feet per Lin. Ft.
'Tween decks No. 1.....	8,070	.....	1,475 }	
'Tween decks No. 2.....	3,530	.....	486 }	5.9
'Tween decks No. 3.....	6,100	*	..... }	
'Tween decks No. 4.....	9,900	.....	1,606 }	5.0
Meat and ice box.....	817	.....	218	3.75
Ship's stores	817	.....	171	4.75
Hold No. 1.....	19,200	15,000	4,477	4.30
Hold No. 2.....	18,500	15,000	4,711	3.92
Hold No. 3.....	19,000	15,500	4,806	3.94
Hold No. 4.....	19,000	14,800	5,264	3.62

* Brine-cooling coil..

The first of the two refrigerating machines—a thirty-ton machine manufactured by the Mayer Ice Machine Company—was installed in New Orleans. This company was also awarded the contract for furnishing and installing all of the brine piping. The compressor is of the horizontal type, with a single steam cylinder 12 inches diameter by 30 inches stroke, fitted with Corliss valve gear. The ammonia cylinder is 10 inches diameter by 22 inches stroke. The condenser for this machine is of the double-pipe flooded type, consisting of 570 feet of 1¼-inch pipe inside of 2-inch pipe. The brine-cooling coils are also double, and are composed of 562 feet of 2-inch pipe inside of 3-inch pipe.

Two duplex brine-circulating pumps, 6 inches by 5¾ inches by 6 inches, a brine-equalizing tank and a duplex circulating pump, 7½ inches by 5 inches by 6 inches, for the condenser complete the refrigerating equipment, which was installed at this port.

The old electric generators were removed and two new 10-kilowatt units installed, with plugs for portable lights in all compartments.

Screwed deck plates were fitted in the main deck over all compartments, and thermometers suspended from them by sash chain extending down through pipe leads. In each hold one pipe terminates below the lower deck, while a second extends to the lower portion, in order that both average and maximum temperatures may be observed.

The price bid on the insulation being considered exorbitant, the work was let out on a cost plus percentage basis

U. S. A. T. McCLELLAN REFRIGERATOR LOG, JUNE 24 TO JULY 3, 1914.

Date.	Hour.	Rev. per Mt.	Pressures.			Temperatures.													
			Brine Pressure.	Condenser Pressure.	Expansion Pressure.	Brine Cooler.	Brine Pump.	Meat Box.	Chill Box.	Tw. 3	Deck 4	1 U	B	2 U	B	3 U	B	4 U	B
June 24	8 p.m.	46	30	195	8	19	26												
" 24	10 p.m.	40	30	190	15	45	54												
" 24	11 p.m.	40	30	190	15	29	38												
" 24	12 p.m.	40	30	190	15	29	38												
" 24	4 a.m.	40	30	190	10	20	32												
" 24	6 a.m.	70	30	190	10	4	12												
" 24	8 a.m.	60	38	205	5	6	14												
" 24	9 a.m.	60	25	205	5	13	19												
" 24	12 a.m.	40	45	190	15	26	34												
" 25	2 a.m.	72	40	195	7½	4	12												
" 25	4 a.m.	70	40	195	7½	3	12												
" 25	12 a.m.	40	35	200	7½	12	18							54	39	34	27		
" 25	1 to 4 p.m.	45	30	190	12	11	18							56	48	32	26		
" 25	4 to 8 p.m.	50	30	190	10	3	30	36	45					54	39	39	27		
" 25	10.30 p.m.	48	40	190	10	13	23							56	46	38	26		
" 25	12 p.m.	72	40	190	10	14	18												
" 26	1 to 4 a.m.	48	40	175	7½	0	6							58	50	36	20		
" 26	4 to 8 a.m.	45	43	175	8	2	6	32	52					58	50	24	19		
" 26	8 to 12 a.m.	60	43	190	14	11	18							52	42	26	22		
" 26	12 to 4 p.m.	45	40	185	13	40	10	33	46										
" 26	4 to 8 p.m.	45	40	182	10	8	14							56	26	26	22		
" 26	8 to 12 p.m.	60	30	180	10	11	14							42	24	26	22		
" 27	12 to 4 a.m.													24	42	22	26		
" 27	4 to 8 a.m.	60	33	190	10	22	28							46	26	26	26		
" 27	8 to 12 a.m.	48	36	182	9	14	20							48	28	32	28		
" 27	12 to 4 p.m.	60	40	185	15	9	18	38	48					46	26	29	22		
" 27	4 to 8 p.m.	56	40	180	8	1	6	38	48					44	22	24	19		
" 27	8 to 12 p.m.	54	40	172	10	0	6							42	18	22	17		
" 28	12 to 4 a.m.	56	40	172	10	3	9							42	18	22	17		
" 28	4 to 8 a.m.	58	40	172	8	2	4							42	16	22	16		
" 28	8 to 12 a.m.	54	40	180	16	14	20	32	48					22	18	24	19	50	50
" 28	12 to 4 p.m.	60	40	180	15	10	16	38	45					24	22	26	22	34	34
" 28	4 to 8 p.m.	56	40	180	13	16	22	38	45					28	24	26	24	32	30
" 28	8 to 12 a.m.	54	42	180	12½	8	14	38	45					24	22	30	22	28	26
" 29	12 to 4 a.m.	54	45	172	13	6	12							26	22	26	20	26	24
" 29	4 to 8 a.m.	54	40	172	12½	7	12	30	45					22	20	25	20	24	22
" 29	8 to 12 a.m.	54	40	160	12	5	10	30	45					22	18	24	18	24	20
" 29	12 to 4 p.m.	58	40	145	12	4	10							20	18	24	18	20	19
" 29	4 to 8 p.m.	56	40	145	10	2	8	30	46					20	18	22	18	19	18
" 29	8 to 12 p.m.	54	45	143	10	2	8	30	46					21	17	20	16	18	16
" 30	12 to 4 a.m.	60	45	140	10	2	8							19	16	22	16	17	16
" 30	4 to 8 a.m.	80	43	140	10	2	8							17	16	20	16	17	16
" 30	8 to 12 a.m.	54	43	132	9	1	6	30	42					17	16	19	15	16	15
" 30	12 to 4 p.m.	58	45	130	10	0	6	30	44					6	14	20	14	16	14
" 30	4 to 8 p.m.	58	43	130	10	2	8	30	42	26	33			17	15	20	15	15	15
" 30	8 to 12 p.m.	72	43	130	9	0	5	30	42	24	26			16	14	18	14	14	12
July 1	12 to 4 a.m.	70	45	130	10	1	6	30	42	20	26			16	14	18	14	14	12
" 1	4 to 8 a.m.	64	43	130	8	0	6	30	44	18	24			15	12	16	12	12	12
" 1	8 to 12 a.m.	64	43	130	10	1 B	5	26	44	17	23			14	12	16	12	12	10
" 1	12 to 4 p.m.	70	45	130	8	1 B	5	26	44	17	21			14	12	12	16	12	21
" 1	4 to 8 p.m.	62	40	130	8	2 B	4			16	20			13	11	14	10	11	10
" 1	8 to 12 p.m.	70	40	125	7	3 B	3			16	18			14	12	14	12	10	8
" 2	12 to 4 a.m.	65	40	125	7½	4 B	2			14	17			12	10	11	9	10	9
" 2	4 to 8 a.m.	70	40	125	7½	4 B	2			14	16			9	8	11	9	9	8
" 2	8 to 12 a.m.	54	40	130	6	4 B	2	24	44	12	15			9	11	9	11	8	8
" 2	12 to 4 p.m.	60	40	130	6	4 B	2	24	44	12	15			11	8	11	8	8	7
" 2	4 to 8 p.m.	60	40	130	5	5 B	1			12	15			11	8	11	9	8	8
" 2	8 to 12 p.m.	56	40	125	6	5 B	1			14	17			11	8	11	10	8	8
" 3	12 to 4 p.m.	54	40	125	5	5 B	0	24	44	12	15			11	8	11	8	7	7
" 3	4 to 8 p.m.	56	40	130	5	6 B	0			10	16			11	7	7	7	6	6
" 3	8 to 12 p.m.	58	40	126	6	5 B	1	26	44	10	14			11	7	10	7	6	6

to H. L. Alexander, a New Orleans contractor, and the entire job of insulating and installing the pipes and machinery was completed in thirty days.

Upon the day of completion, the *McClellan* was ordered to New York City, where a second machine—a York 20-ton compressor—was installed; a cargo of beef was shipped, and the vessel left immediately for the Gulf of Mexico, where she has since been in continuous service. The table shows a log of the try-out on the run between New Orleans and New York.

Care of Boilers

When boilers are under steam irregulatr working should be guarded against as much as possible. The water in the boiler should occasionally be tested for acidity, which can be easily done by means of a slip of litmus paper dipped in the salinometer. In cases where water losses are necessarily made up from the sea, blowing, except for the purpose of scumming, should not commence until a saturation of from 3 to 3½ is reached, the surface blow being preferably used for this purpose when steaming, and the bottom blow when under low-banked fires. When through steaming no haste should be used in reducing the boiler pressure, but the steam should be allowed to work off gradually through bleeders or evaporators, the dampers should be nearly closed, only leaving enough open to allow the escape of gas, and furnace doors open. After fires are hauled care should be taken to wet them down only after being removed from the boiler front. An excellent protection is afforded by a light plate of iron fitted in front of the ash pits. The furnaces should finally be closed and the boilers allowed to cool gradually. In case it is necessary to blow the boiler down instead of pumping it out when cold, the pressure should be allowed to drop to about 20 pounds, depending upon circumstances, thereby avoiding the severe jar liable with higher pressures. In case of jar it may frequently be checked by either opening or closing the blow valve slightly.

Steam gages on boilers should be tested at intervals to see that they are adjusted correctly, as it frequently happens that gages on different boilers, all connected up, show a variance of from 1 to 10 pounds.

Costly Congestion at Marine Terminals

Antiquated Facilities and Poor Delivery Systems Boost Costs of Handling Freight at Docks—Remedies Suggested

BY A. JACKSON MARSHALL*

Freight congestion at the terminals and steamship piers of all our large cities, and especially New York, has become a very knotty problem during the past year, and we are hearing a great deal about the chaotic condition into which freight distribution, both domestic and foreign, has fallen. The reason for the unusual congestion of marine terminals just at this time is, of course, obvious when we consider that approximately 1,000 vessels, of which a large proportion were merchant steamers, with an estimated tonnage of 2,000,000 tons, have been destroyed, directly or indirectly, as a result of the European war, while in addition to this are the great numbers ot vessels which have been taken for transport service or have been interned. At present over one million tons of freight are stalled at New York terminals, and steamship piers in all of our seaport cities are stacked to the roof beams with freight intended for distribution. The shortage of ocean tonnage makes it difficult to take care of even the ordinary trade, while the added facilities demanded by bumper crops, the movement of war material and increased exports are greater than present conditions can supply, and the result is this great accumulation of freight at marine terminals.

Delay of Vessels in Port

While the shortage of ocean carriers is the major reason for present conditions, a great deal of the congestion would be considerably relieved if freight were handled in a more efficient manner at the terminals. Because of old-fashioned methods, antiquated handling facilities and confused schedules, steamers are delayed for long periods. Vessels can be rapidly loaded if cargoes are delivered properly and if they can pass through the terminals with a minimum amount of delay. With the present lack of system and inefficient handling of freight this is utterly impossible. Mr. C. H. Betts, manager of the Australian department of French, Eddye & Company, regards the shortage of pier facilities and the poor methods of moving freight in terminals as the most serious problem in connection with the commerce of our harbors. The export freight situation has become so serious that some cargoes, which should have been moved over two months ago, have not yet been tendered for shipment.

Not only has export trade been affected by the congested condition of our terminals, but domestic transportation suffers as well. Shortage of ocean tonnage is not responsible for the fact that local merchants are unable to get their goods through the terminals. It is estimated that there is a total of 720,000,000 tons of goods every year that passes through the railroad freight and transfer stations of this country, and 600,000,000 tons handled at marine terminals, making a grand total of 1,320,000,000 tons of freight handled annually in the United States, or 4,400,000 tons every working day in the year. The present lack of system in handling this freight and the tremendous amount of lost time involved represents a yearly loss of more than $80,000,000 (£16,400,000) at railway terminals alone. We pay $2,000,000,000 (£410,000,000) a year in the United States for freight transportation, four-fifths of which, or about $1,600,000,000 (£328,000,000), is for terminal work.

The fact that drayage and terminal expenses of freight transportation are 22 percent more than the line haulage of the same freight over 1,000 miles of railroad is a condition toward which every effort should be exerted to remedy. No one thing alone can solve the problem. In the first place, co-operation between railroads, steamship companies, trucking concerns, manufacturers and the public itself is vital. Certain it is that, as surely as it is a problem in transportation, improved transportation appliances, when once established through the united efforts of everyone interested, will work its solution. In the forerank among these appliances stand the commercial vehicle, the electric industrial truck, overhead telpherage systems of freight movement, the gravity conveyor, etc. As the connecting link between the terminal and the merchant, the motor truck must assume the first position among transportation facilities.

Savings with "Electric Stevedores"

In places where some modern equipment has been installed, such as the electric industrial truck, a more systematic condition has resulted and a considerable economy been effected. In the railway terminals where the electrically propelled baggage truck has supplanted the hand truck, it is estimated that there is a total saving of 66 percent. Down in Savannah, where the "electric stevedore" has been in use for some time, you may see cotton moved across great piers by these little electric trucks equipped with hoists and cranes, carrying it directly into the holds of the steamers which bring it North. Two of these electric industrial trucks would handle more cotton in one day than twenty freight handlers with hand trucks.

Under the method of hand-truck operation, the cost per ton of handling freight at railroad terminals and steamship piers is 25 cents (1/0½), while the cost per ton for performing the same work with electric industrial trucks is but 10⅓ cents (0/5 1/6).

As the use of the "electric stevedore" increases and other modern methods and appliances are installed it is to be hoped that this will influence a further systematized manner of delivery. As this work becomes more thoroughly unified, freight sheds and docks may find it desirable to operate on a full or partial 24-hour schedule. In cases where the limited ground space available in our city terminals prevents the installation of mechanical carriers that would bring about economy and efficiency, the remedy which appears most feasible to the writer is the establishing of large outer platforms or warehouses where all outbound package freight could be assembled and consolidated, and where the floor movement would be performed by the electric industrial trucks. If the co-ordination of the railroad and highway movement of merchandise and a collaboration of the railroads and teaming interests could be effected, there is every reason to believe that a powerful and efficient trucking organization could be formed which would effect not only an enormous saving over present cost, but render greatly improved service through the employment of scientific and efficient methods of operation and a truck-despatching system worked out along the same lines as is the train-dispatching and car-distribution systems of our railroads.

* Secretary Electric Vehicle Association of America, 29 West Thirty-ninth street, New York City.

It is of interest to realize that the railroad and steamship companies, in adopting or promoting some such scheme, would in all probability use electric trucks as the transportation medium. Since much freight would be conveyed at night, either to the big wholesale houses or to well-located centers, from which points goods could be advantageously distributed the following day, a vehicle operating as noiselessly as possible would be required. Owing to its practically silent operation, the electric vehicle would be ideal for traversing city streets at night, since they would not be obtrusive in passing through the residential districts. Furthermore, the recognized economy of the electric truck for city use and its absolute dependability are points in its favor for a service of this kind. Electrics are free from the unsanitary and objectionable features attached to the horse and gasoline (petrol) vehicles, and it is interesting to note that, while gasoline is expensive and the price is still soaring, electricity is constantly becoming cheaper. Moreover, there is not associated with the electric the ever-present fire hazard of the gasoline machine.

An efficient and well-regulated trucking organization would, without doubt, establish central clearing stations such as have been described, at convenient points in the business districts, to which light-capacity vehicles could be run from nearby warehouses, and from which consolidated loads could be moved in high-capacity trucks to the railroad and steamship terminals, bringing back on their return similar heavy loads of inward freight for redistribution from the central clearing house. Thus there would be secured substantially an extension of the railroad and steamship operation methods beyond the terminals. Warehouse to warehouse delivery is successfully carried on in Europe, and from the fact that over 60 percent of freight haulage in our large cities is done under contract with truckmen, it is evident that the business world in general would welcome an improvement along these lines. Confusion or congestion in freight terminals reacts throughout the entire transportation line using that terminal, hence it is safe to suppose that the solution of terminal freight congestion lies along the lines of improved street transportation by the trucking industries, through consolidation and through the use of an efficient, modern carrier—the motor truck. But, in order to secure maximum efficiency, order must be brought out of chaos, and the delivering, handling and receiving of freight at terminals must be organized at least as systematically as the railroads and steamship companies handle freight while in transit. The high point of efficiency to which the steamship companies and railroads have developed their systems of line transportation stands out prominently and most favorably when contrasted with the unorganized and disordered conditions at present existing in terminal freight handling. A development of such far-reaching influence must sooner or later attract the best minds—then we can hope for action.

Freeboard: its Preliminary Determination

Type of Ship—Dimensions—Mean Sheer—Round of Beam—Description of Deck—Coefficient of Fineness

BY H. A. EVERETT *

Freeboard is the term given to the height that the vessel's deck at the side is above the surface of the water. In the early days of steamships, overloading was common, and in order to prevent the dangerous overloading of vessels there were early legislative acts which required that certain marks should be plainly shown on the sides of the vessels below which it should be illegal to immerse the boat.

Up to 1876 there had been practically no legislative action, but at about this time the British Parliament required a mark as above stated, which, however, was affixed by the owners, and its location was dependent entirely upon the owner's judgment. For this reason the mark had but slight effect in minimizing the evil and in 1882 Lloyds issued tables in which they carefully worked out proper freeboards for different vessels which were largely adopted by the owners. The following year the British Board of Trade appointed a committee to investigate the question thoroughly, and this committee eventually drew up tables which were ultimately embodied in the Merchant Shipping Act of 1890, which may be considered as the basis of the present practice. This act is still in effect, though slightly modified in 1906 and 1909.

The basic principle of these rules is that it is desirable for a vessel to be loaded no deeper than will leave a margin or reserve of buoyancy which shall be sufficient quickly to raise the boat over seas rather than to allow the seas to come on board—in other words, that the vessel shall be sufficiently lively to be seaworthy. Indirectly, the freeboard is associated with structural strength, as a vessel which is deficient in strength should not be permitted to load as deeply and encounter such severe stresses as one which is stronger structurally. The ability of a ship to rise and not take on water is a function of the amount and location of her reserve of buoyancy.

In assigning freeboard, vessels are divided into four classes:

1. Full scantling steamers.
2. Spar deck steamers.
3. Awning deck steamers.
4. Sailing vessels.

Since the revision of Lloyds' "Rules for the Construction of Steel Vessels" in 1909, this organization has officially sanctioned by two classes of steamers, viz., full scantling and shelter deck, but the freeboard rules still hold to the classes given above with modifications for the shelter-deckers, as will be shown later. The full scantling vessel is taken as of sufficient strength, and is the standard by which strength is gaged. Vessels which are less strong are required to have more freeboard. For the full scantling vessel the freeboard is determined solely by the desirable reserve of buoyancy.

For a given ship belonging to either of the classes 1 and 3, the exact freeboard required may be calculated by constructing a displacement scale to the height of the deck to which the freeboard is measured, so as to give the whole molded volume up to the upper surface of that deck. The percentage of the total volume which is given on Plate I as a reserve of buoyancy for a vessel of given type and dimensions will be the amount of volume that must be

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left out of the water. If a line be drawn upon this displacement curve at a draft sufficient to cut off the given percentage of total volume, the height of side above this draft will be the freeboard required. (See Fig. 1.)

In order to simplify and reduce the work that would be involved by the above mode of determining the maximum allowable draft and the consequent freeboard that corresponds to a given percentage of reserve buoyancy, tables were evolved which, for a ship that conformed to a so-called standard ship, gave directly the percentages of reserve of buoyancy and freeboards necessary for different sizes and types of vessel. The curves of Plates II and III are plotted from these tables.

The *"standard ship"* was considered to be a flush deck ship with a certain sheer which was termed "standard or normal sheer," with a certain proportion of length to depth

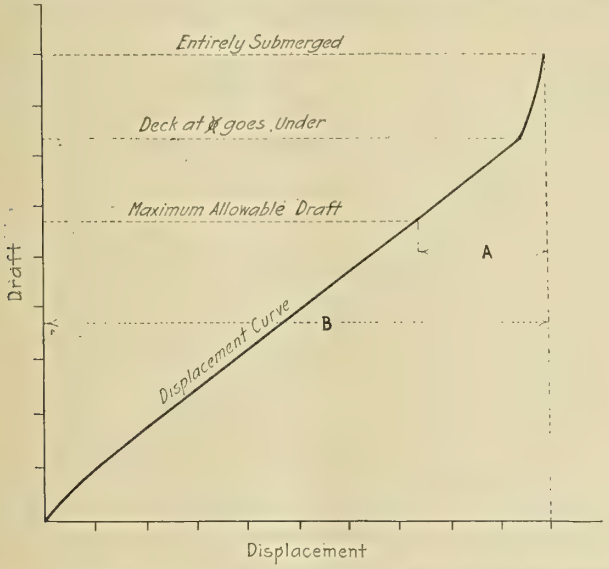


Fig. 1.—When A is the Percentage of B given by the curves on Plate I, then the Draft is the Maximum Permitted and Depth—Draft=Freeboard

and with a "standard" roundup of deck beams, and for this vessel the curves read directly. Deck erections, of course, contribute to safety and are taken account of as a corrective term, as are also other variations from the standard ship. In practice the freeboard is actually assigned to the ship after the ship is built and usually by one of the classification societies' agents, but its preliminary determination is an important and necessary item in the design of any vessel, as the draft plus the freeboard gives the depth.

The complete tables as issued by the British Board of Trade take up all of the possible modifications and corrections which are involved by vessels differing from the arbitrarily assumed standard. The following work is based upon the rules directly, although the presentation and the wording are modified to a large extent. The curves given on Plates I-IV are a graphical representation of corresponding tables in the rules. Spar deck steamers and sailing vessels are not included, as those classes are not numerous in present-day new designs.*

The limitations of loading, as laid down by the above mentioned act, are represented by a disk and number of horizontal lines called "maximum load lines" which are cut and painted on the side of the ship amidships, as shown by Fig. 2. The upper edge of each line is the point of measurement.

* For further information in this respect and for details of other elements not dealt with here, refer to the publication of the Marine Department of the British Board of Trade, issued in May, 1909, entitled "Instructions to Surveyors, Load Line," available at Wyman & Sons, Ltd., Fetter Lane, London, E. C.

The word freeboard, legally, denotes the height of a side of the ship above the waterline, measured at the middle of her length along the load waterline. It is measured from the top of the deck at the side. The reserve of buoyancy necessary for flush deck steamers of full scantling and awning deck steamers are given by the curves on Plate I, and these curves hold for any and every vessel, regardless of proportions. For the *standard* vessel of these classes and within the dimensions given the *freeboards* required may be *read directly* from Plates II and III.

For awning deck vessels the freeboards are determined more by considerations of structural strength than by reserve of buoyancy, and indicate the depth of loading beyond which it is probable that first-class vessels of this type would be unduly stressed when at sea. Therefore the freeboards and percentages of reserve buoyancy are

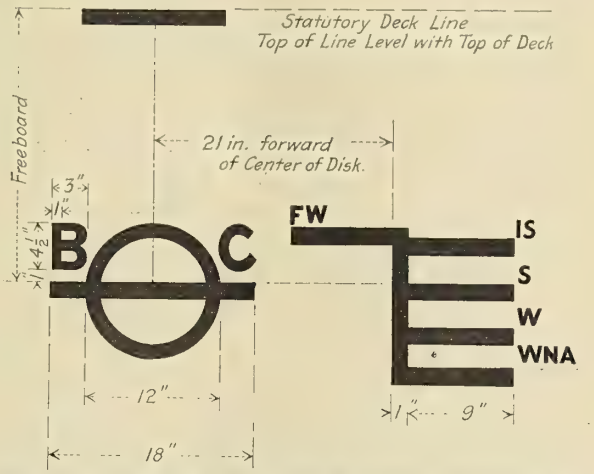


Fig. 2.—The Letters B. C. signify British Corporation*; F. W., Fresh Water Allowance; I. S., Freeboard in the Indian Seas in Summer; S., Summer Freeboard; W., Winter Freeboard; W. N. A., Winter, North Atlantic Freeboard

in *excess* of what would be required for full scantling vessels. They are measured to the *deck below* the shelter or awning deck. The freeboards given in the curves are for flush deck vessels in all cases and for the standard ship—a ship which has no deck erections, has a proportion of length to depth of 12, has a roundup of deck beams of ¼ inch per foot of beam, and has a mean sheer in accordance with that derived from the curves shown on Plate IV.

The data which are required for determining the freeboard by the curves are:

1. Type of ship.
2. Dimensions.
3. Mean sheer.
4. Round of beam.
5. Description of deck (where statutory deck line is placed).
6. Coefficient of fineness.

The *type* of ship must be agreed upon in order to ascertain which table will meet the case and whether modifications are necessary.

The *length* for freeboard of the vessel is measured at the load line from the fore side of the stem to the after side of the sternpost in sailing ships, and the after post in steamers.

The *breadth* for freeboard is the extreme breadth measured to the outside of plank or plating as given on the certificate of the ship's registry.

The *depth* for freeboard is the depth of hold as given

* Or L. R. for Lloyds Register, and B. V. for Bureau Veritas.

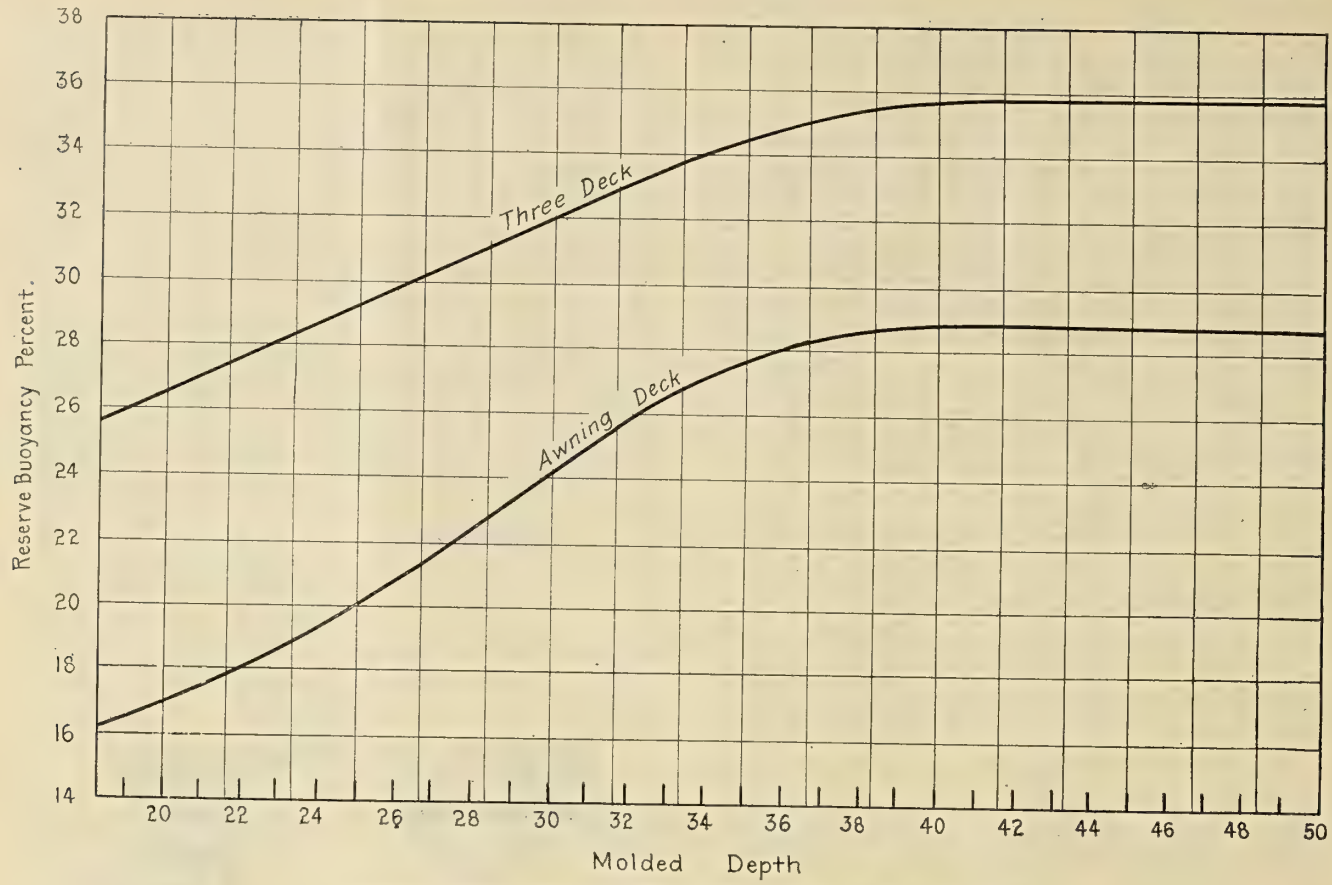


Plate I.—Reserve Buoyancy (Winter) Required by the Board of Trade

on the certificate of the ship's registry. This is the depth used for determining the coefficient of fineness. (Upper deck beam at side in flush deck vessel, main deck beam at side in spar and awning deck vessels to top of ceiling or sheathing on double bottom.)

Coefficient of Fineness

The coefficient of fineness in one, two and three-deck vessels is found by dividing one hundred times the gross registered tonnage of the vessel below the upper deck by the product of the length, breadth and depth of hold. In shelter deck vessels the registered depth and tonnage are taken to the deck below the shelter deck.

Molded Depth

The molded depth of an iron or steel vessel, as used in the curves, is the perpendicular depth taken from the top of the upper deck beam at side, at the middle of the length of the vessel, to the top of the keel and the bottom of the frame at the middle line. This is the depth which is used for the proportion of length to depth.

Freeboard

The molded depth, taken as above described, is that used in the curves for ascertaining the amount of reserve buoyancy and corresponding freeboard in vessels having a wooden deck, and the freeboard is measured from the top of the wood deck at side, at the middle of the length of the vessel. Where wood decks are not fitted on the upper decks, the freeboard should be reduced by the thickness of the wood deck or the percentage of it corresponding to the percentage of the length covered by substantial deck erections if they cover less than 70 percent.

The following example will illustrate the application of the curves when dealing with a "standard" vessel:

In a steamer of the following dimensions, viz.: length, 357 feet; extreme breadth, 40 feet; depth of hold, 26 feet; registered tonnage under deck, 2,980 tons, and *molded* depth, 29.8 feet, the under deck capacity in cubic feet is 298,000; by dividing this by 382,000—that is, the product of the length, breadth and depth of hold—the quotient is 0.78, or the coefficient of fineness.

If we now refer to Plate II at 29.75 feet molded depth and coefficient 0.78, the winter freeboard given for a *standard steam vessel* (without erections and length twelve times the molded depth) is 7 feet 7 inches, which corresponds to a reserve buoyancy of 32 percent of the total bulk.

It is obvious from the foregoing that the determination of the freeboard for a standard vessel is extremely simple. As a matter of fact, however, vessels rarely conform to the proportions assumed for the "standard," and the correct determination of freeboard for the actual vessel becomes a matter of properly applying the corrections to allow for the departure from the standard. The variations most commonly met with are in the sheer, deck erection and proportions of length to depth. The corrections for each of these items must be made and in the order given, as the correction for erections is based upon the difference between the freeboard for full scantling vessels *corrected for sheer* and the freeboard for awning deck vessels (uncorrected).

Sheer

The tables are framed for vessels having a mean sheer of deck *measured at the side,* as shown in the sheer diagram of Plate IV.

In flush-deck vessels and in vessels with erections on deck, when the sheer of deck is greater or less than the above, *and is of a gradual character,* divide the difference in inches between it and the mean sheer provided for by 4, and the result in inches is the amount by which the

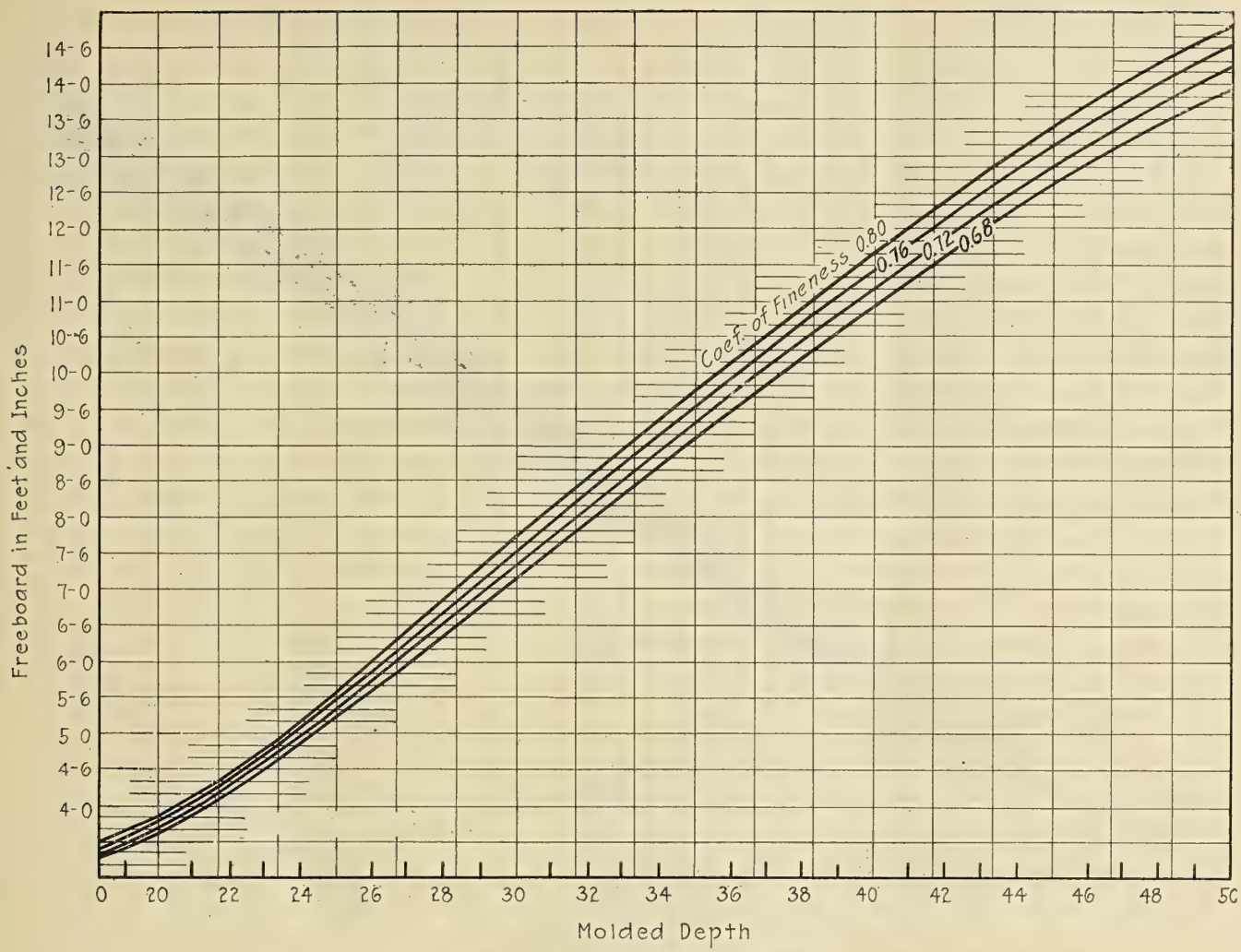


Plate II.—Minimum Freeboard (Winter) Allowed by the Board of Trade for Three Deck Steamers

freeboard amidships should be diminished or increased, according as the sheer is greater or less.

In all cases the rise in sheer forward and aft is measured with reference to the deck at the middle of the length, and where the lowest point of the sheer is abaft the middle of the length, one-half of the difference between the sheer amidships and the lowest point should be added to the freeboard specified in the tables for flush-deck vessels and for vessels having short poops and forecastles only.

Erections on Deck

For steam vessels with topgallant forecastles having long poops, or raised quarter decks *connected* with bridge houses, covering in the engine and boiler openings, the latter being entered from the top and having an efficiently constructed iron bulkhead at the fore end, a deduction may be made from the freeboard given in the curves, according to curve *A*, Plate IV.

When the erections on a vessel consist of a topgallant forecastle, a short poop having an efficient bulkhead and bridge house *disconnected,* the latter in steamers covering the engine and boiler openings, and being efficiently enclosed with an iron bulkhead at each end, a deduction may be made from the freeboard given in the curves, according to curve *B*, Plate IV.

When the erections on a vessel consist of a topgallant forecastle and bridge house only, the latter in steamers covering the engine and boiler openings, and being efficiently enclosed with an iron bulkhead at each end, a deduction may be made from the freeboard given in the curves according to curve *C*, Plate IV.

When the erections on a steam vessel consist of a short poop or raised quarter-deck of a height from 3 feet to 6 feet for lengths of ship of 250 feet to 400 feet, and topgallant forecastle only, the former being enclosed at the fore end with an efficient bulkhead, and when the engine and boiler openings are entirely covered, a deduction may be made from the freeboard given in the curves according to curve *D*, Plate IV.

Vessels of Extreme Proportions

For vessels whose length is greater or less than twelve times the molded depth for which the curves are framed, the freeboard should be increased or diminished as specified in the following table. Thus, if the vessel in the above example were 367 feet long, the winter freeboard

TABLE I.

Correction in Insurance for a Change of 10 Feet in Length.	Molded Depth, Feet.	Length, Feet.
1.2	20—23	240—276
1.3	23½—25½	282—306
1.4	26—28	312—336
1.5	28½—30½	342—366
1.6	31—33	372—396
1.7	33½—50	402—600

For shelter deck vessels the correction is one-half that specified in the above table.

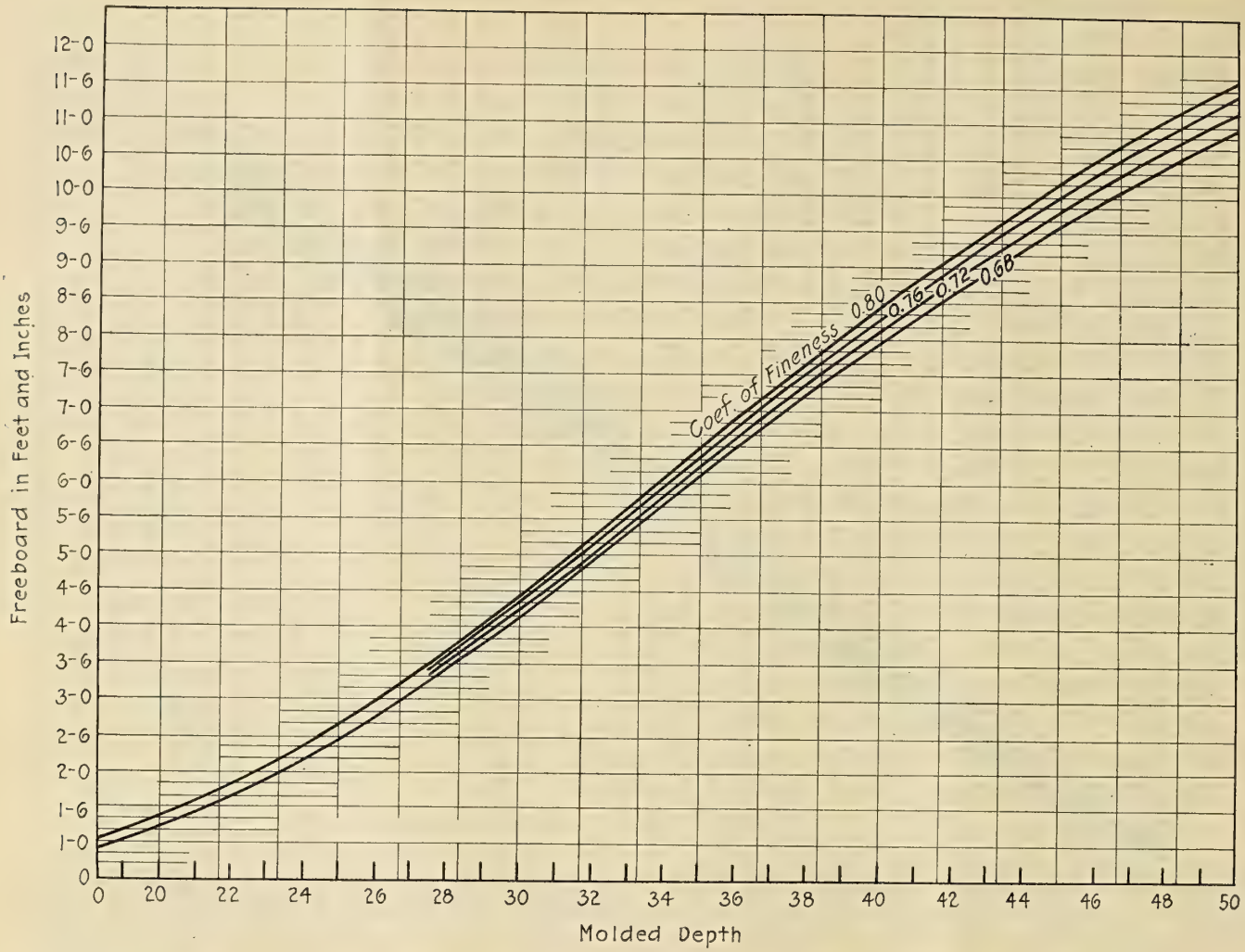


Plate III.—Minimum Freeboard (Winter) Allowed by the Board of Trade for Awning Deck Steamers

required would be 7 feet 7 inches plus 1.5 inches, or 7 feet 8.5 inches. For steam vessels with normal enclosed deck erections, as on Plate IV (curve *A* and *B*), extending over 6/10 or more of the length of the vessel, the correction for length should be one-half that specified in the plot.

Round of Beam

In calculating the reserve of buoyancy, an allowance has been made of ¼ inch for every foot of the length of the midship beam for the roundup. When the round of the beam in flush-decked vessels is greater or less than given by this rule, divide the difference in inches by 2 and diminish or increase the freeboard by this amount. For vessels with erections on deck the amount of the allowance should depend on the extent of the main deck uncovered. This rule for round of beam does not apply to shelter or awning deck vessels.

Breadth and Depth

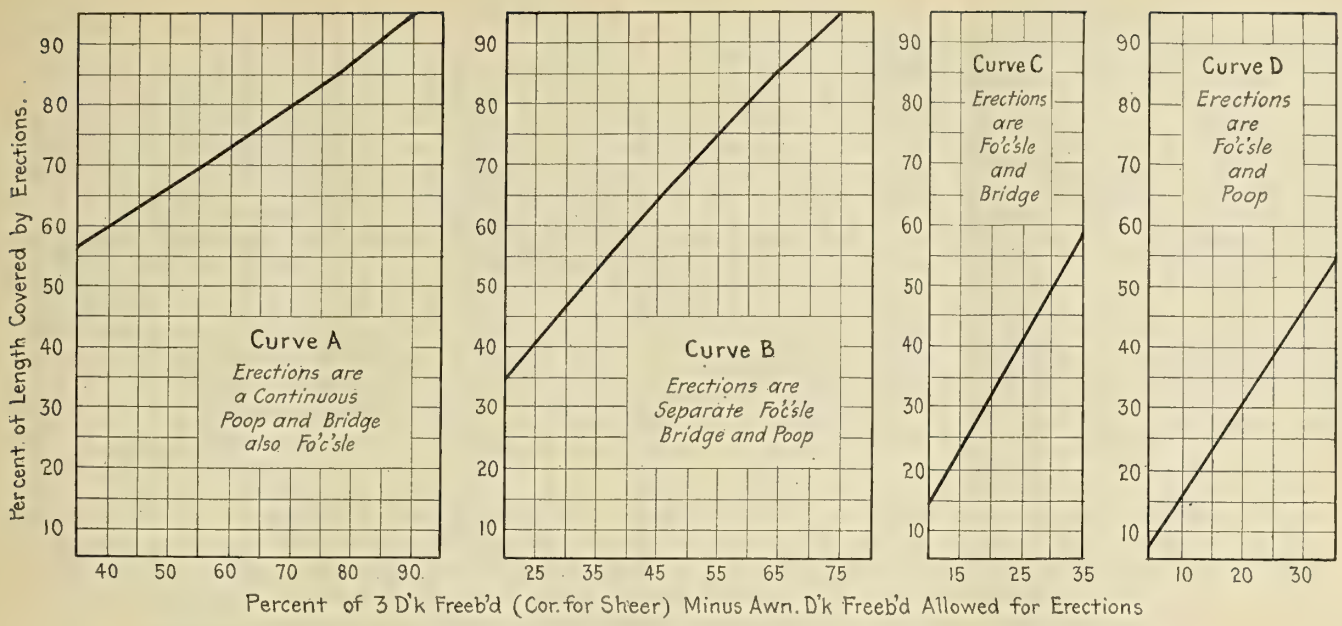
It has been assumed that the relation between the breadth and depth is reasonable, and for vessels of less relative breadth the freeboard should be increased to provide a sufficient range of stability. The following example will illustrate the application of the curves when dealing with a vessel not conforming to the standard type:

A vessel is 234 feet long, 29 feet broad and has a molded depth of 17 feet, the coefficient of fineness being .72. Suppose the vessel to have a poop and bridge house of the united length of 121 feet and a forecastle 20 feet in length, and let the sheer forward, measured at the side, be 4 feet 6 inches, and aft 2 feet 1 inch.

	Ft.	In.
Freeboard by Plate II, if of the standard proportions, without erections, and with the normal amount of sheer	2	11
The mean sheer by rule is 33.4 inches, or 6 inches less than that in the vessel, and the reduction in freeboard is 6 inches divided by 4	0	1½
Freeboard of vessel without erections and with 39½ inches mean sheer	2	9½
Freeboard by Plate III as awning-decked	0	9½
Difference	2	0

The combined length of the erections is 141/234, or six-tenths of the length of the vessel, and the allowance for erections from curve *A*, Plate IV, will be, therefore, four-tenths of 24 inches, or 9½ inches. We have therefore

	Deduct Inches
Amount deducted from freeboard for excess of sheer	1½
Amount deducted from the freeboard for erections	9½
Amount deducted if vessel be fitted with an uncovered iron main deck = 6/10 × 3½	2
	13
The length being 30 feet in excess of that for which the tables are framed, the addition to the freeboard in respect of the same is one-half of 30/10 of 1.1 inches, or	1½
	11½


Curve A
Erections are a Continuous Poop and Bridge also Fo'c'sle
Curve B
Erections are Separate Fo'c'sle Bridge and Poop
Curve C
Erections are Fo'c'sle and Bridge
Curve D
Erections are Fo'c'sle and Poop
Per cent of Length Covered by Erections.
Percent of 3 D'k Freeb'd (Cor. for Sheer) Minus Awn. D'k Freeb'd Allowed for Erections

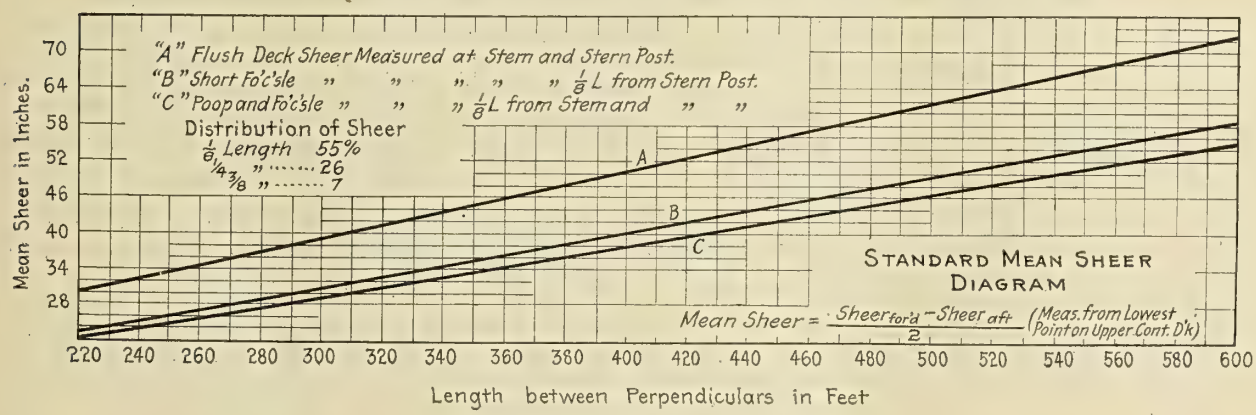

"A" Flush Deck Sheer Measured at Stern and Stern Post.
"B" Short Fo'c'sle " " " " " 1/6 L from Stern Post.
"C" Poop and Fo'c'sle " " " 1/6 L from Stem and " "
Distribution of Sheer
1/6 Length 55%
1/4 " 26
3/8 " 7
A
B
C
STANDARD MEAN SHEER DIAGRAM
Mean Sheer = Sheer for'd − Sheer aft / 2 (Meas. from Lowest Point on Upper Cont. D'k)
Mean Sheer in Inches.
Length between Perpendiculars in Feet

Plate IV

That is 11½ inches to be deducted from 2 feet 11 inches, leaving a winter freeboard of 1 foot 11½ inches.

Corresponding summer freeboard, 1 foot 9 inches.

Vessels loaded in fresh water may have less freeboard than that given in the several tables, according to the following scale:

TABLE II.

Molded Depth, in Feet.	Reduction in Freeboard.	
	Vessels Without Erections on Deck.	Shelter and Awning Deck Vessels.
19 and under 22	4	4½
22 " " 25	4½	5
25 " " 28	5	5½
28 " " 31	5½	6
31 " " 34	6	6½

Note.—The weight of a cubic foot of salt water is taken, in the above table, to be 64 pounds and that of fresh water 62.5 pounds.

In no case shall the deepest loadline in salt water, whether indicating the summer or Indian summer line, be assigned at a higher position than the intersection of the top of the upper deck with the vessel's side, at the *lowest* part of the deck.

In the case of shelter-decked vessels the deck next below the shelter deck is to be regarded as the upper deck.

So far the question of freeboard determination has been considered from the viewpoint of its determination for some existing ship of which the characteristics are known. The most useful function of the work as presented here, however, is to permit a solution for the depth of a vessel under design. The *accurate* determination of freeboard should properly be attempted only from the complete tables referred to earlier, but from the information here presented, it is possible to readily determine the freeboard and therefore depth for a proposed design which has progressed sufficiently to have its length, draft, block coefficient and general arrangement selected. In general the coefficient of fineness" is sufficiently close to the block coefficient to accept the latter for entering the curves.

In considering a vessel under design, then, the general procedure for determining the freeboard of a full scantling vessel should be as follows:

1. Assume a *molded depth* which seems reasonable, enter the curves, and for this *depth* read off the freeboard for the proper coefficient of fineness.

2. Correct this freeboard for sheer and erections and add the corrected freeboard to the draft to determine a revised molded depth. Multiply it by 12 and the difference between this and the actual length gives the basis for determining the corrections for proportions.

3. Determine the correction for proportions and the original freeboard corrected for these three elements (sheer, erections and proportions) when added to the draft should give a molded depth in agreement with that originally as-

sumed. If it does not, repeat the solution, starting with a modified assumed depth. The first trial will rarely give agreement, but the second or third should suffice, or a curve for interpolation may be plotted as in Fig. 3.

The shelter deck steamers now form such a large proportional part of the tonnage afloat that they need to be built with as a special class, and the revised rules do so take cognizance of them. The freeboard of a shelter deck steamer must in no case be less than the freeboard which would be assigned to a complete awning deck steamer of the same dimensions. The shelter deck rules are framed for a vessel having a complete superstructure covering the full length of the vessel, the deck continuous and unbroken at the side, but having one or more openings along the middle line of deck, such openings not to have *permanent* means of closing in the shape of hatchways fitted with

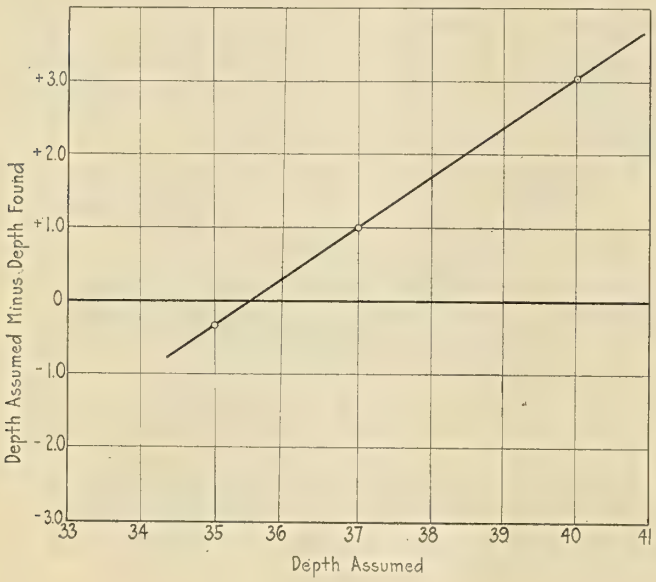


Fig. 3

coamings, cleats, etc. The deck below the shelter deck is called the upper deck and is the one to which freeboard is measured.

For shelter deck vessels the steps for determination of freeboard are the same as in full scantling vessels, considering them as full scantling vessels with very long erections and the freeboard is measured to the deck below the shelter deck (upper deck). The order of procedure is:

1. Assume a reasonable depth (molded) and read the freeboard from the curve for this abscissa on Plate II.

2. Correct this for sheer and use this corrected freeboard in estimating allowances for erections.

3. Correct for erections.

4. Use this newly corrected freeboard for determining the depth (molded) for proportions. Multiply it by 12 and

5. Correct for proportions.

6. Add this final freeboard to the draft and get a depth which should agree with that first assumed. If it does not, repeat the solution. Two or three trials should suffice.

In assigning freeboards to shelter-decked vessels, the following rules should be observed:

1. In making the sheer correction in accordance with paragraph on "sheer," the sheer is to be measured at the ends of the vessel, and the freeboard corrected for sheer is to be used in estimating the allowance for erections.

2. (*a*) If there is but one opening in the shelter deck the allowance for deck erections is to be determined from curve *A*, Plate IV, provided that the effective length of the shelter deck is not less than six-tenths of the length of the vessel.

(*b*) If there are two or more openings in the shelter deck the allowance for deck erections is to be determined from curve *B*, Plate IV, provided that the effective length of the shelter deck, excluding openings, is not less than six-tenths of the length of the vessel.

3. The effective length of the shelter deck is to be calculated in the following manner, provided the openings in the shelter deck do not exceed half the vessel's breadth at the middle of the length of the opening. The length is to be taken as if each opening were an open well. The value of each part is to be assessed in accordance with the different regulations affecting poops, bridge houses and forecastles, open or closed. The final allowance for erections will depend upon whether or not temporary but efficient means are provided for closing the openings in the shelter deck.

(*a*) If *efficient means* as specified below *are provided* for temporarily closing the openings in the shelter deck, the effective length of the shelter deck is to be reckoned as the length computed as prescribed above, plus half the difference between that length and the length of the vessel.

(*b*) If *efficient means* for temporarily closing the openinge *are not provided,* the effective length of the erections is to be computed by adding to the length computed as above one-fourth, instead of one-half, the difference between that length and the length of the vessel.

(*c*) If the openings in the shelter deck are wider than the half beam at that point, the addition to the assumed length of erections is to be modified in proportion to the relation which the actual opening holds to the specified breadth and to a complete well.

To illustrate the method of determining the depth for a new design, and also the application of the rules to the shelter deck type of vessel, the following example will be useful.

A complete shelter-deck vessel 490 feet long by 58 feet beam and 28 feet draft, with one tonnage opening in shelter deck. Block coefficient = .80. Assume for the first trial depth

$$\frac{L}{12}, \text{ or } \frac{490}{12} = 40 \text{ feet, approximately.}$$

	Feet
At 40 feet depth and .80 coefficient of fineness the freeboard for a full scantling vessel is 11 feet 8 inches (Plate II)	11.67
The sheer forward is 9 feet and aft 3 feet, so the mean sheer is $\frac{(9+3)}{2} \times 12 = 72$ inches. The standard or normal mean sheer from Plate IV is 60 inches, so that the excess sheer is $72 - 60 = 12$ inches, and the sheer correction is $12 \div 4 = 3$ inches $= .25$ foot	.25
This is to be subtracted, as the sheer is greater than the normal, then freeboard corrected for sheer is	11.42
Freeboard for awning deck (Plate III) (uncorrected)	8.41
Difference	3.01

The correction for erections is 90 percent of this (curve *A*, Plate IV) as the erections cover 95 percent of the vessel's length, $0.9 \times 3.01 = 2.71$. The freeboard corrected for sheer and erections then becomes $11.42 - 2.71 = 8.71$ feet; this, with a draft of 28 feet, gives a depth (molded) of $8.71 + 28 = 36.71$ feet.

A "standard" ship of this depth would have a length twelve times as great, or $36.71 \times 12 = 440$ feet (approximately), which is 50 feet shorter than the actual ship, so the freeboard must be increased to correct for proportions. From Table I this correction $\frac{1.7}{2}$ inch for every

10 feet excess of length, and the correction in feet is

$$\frac{1.7}{2} \times \frac{1}{12} \times \frac{50}{10} = .0708 \times \frac{50}{10} = .35 \text{ foot.}$$

Therefore the freeboard corrected for sheer, erections and proportions becomes 8.71 + .35 = 9.06 feet, and the depth (molded) is 9.06 + 28 = 37.06 feet.

This depth does not agree with that first assumed, so a second solution will be made, using the depth just found as a trial depth. Assume for the second solution a trial depth of 37 feet.

Freeboard of full scantling vessel	10.50
Less sheer correction	.25
	10.25
Freeboard for awning deck vessel	7.25
Difference	2.99
Correction for erections = .9 × 2.99	2.69
Freeboard corrected for sheer and erections	7.56
Draft	28.00
Depth of ship at 28 feet draft	35.56
Corresponding length of standard ship	427.00
Length of actual ship	490.00
Difference	63.00
Correction for proportions (.0708 × 63)	.43
Final corrected freeboard 7.56 + .43	7.99
Depth at 28 feet draft = 28 + 7.99	35.99

Repeating this process for a third trial depth of 35 feet, we obtain a resulting depth of 35.31 feet and freeboard of 7.31 feet.

From the above three solutions a plot like that of Fig. 3 may be made and the correct depth and freeboard determined from it for the case in hand. These are depth = 35.5 feet and freeboard (winter) = 7 feet 6 inches, or a summer freeboard of 7 feet.

TABLE III

Allowable Reduction from Winter Freeboard for Summer Freeboard. Double these reductions allowed for I. S. line and 2 inches more required for W. N. A. line if of 330 feet length or less.

Molded Depth.	Reduction, Inches.	Molded Depth.	Reduction, Inches.
16′ 6″ to 19′	2	34′ to 35′ 6″	6
19′ to 22′	2½	36′ to 37′ 6″	6½
22′ 6″ to 24′ 6″	3	38′ to 39′ 6″	7
25′ to 26′ 6″	3½	40′ to 41′ 6″	7½
27′ to 28′ 6″	4	42′ to 43′ 6″	8
29′ to 30′ 0″	4½	44′ to 45′ 6″	8½
30′ 6″ to 32′ 0″	5	46′ to 47′ 6″	9
32′ 6″ to 33′ 6″	5½	48′ to 50′	9½

Modern Submarines in War and Peace—IX

Submarines for the Recovery of Shellfish—Future Development of the Submarine and its Use in Warfare

BY SIMON LAKE

A submarine vessel of the Lake bottom-working type has been designed and is about to be built for the location of and the recovery of shellfish on a large scale. Shellfish abound on both the Atlantic and Pacific coasts in great quantities. They are about the most delicious and nutritious food known to man. The most common shellfish are the oyster, the round or hard-shell clam, the long-neck or soft-shell clam, the scallop, and on the Pacific coast, the abalone, which is valuable both for its mother-of-pearl shell, and its meat, which is a great delicacy, the most of which is sent to Japan either dried or canned.

My own sea-bottom investigations, combined with the sea-bottom investigations of the United States Fish Commission, have led me to the conclusion that edible shellfish abound in such great quantities along our coast that they can become an important rival to our meat-growing and packing industries, provided the proper apparatus is used for their recovery. I have, when "wheeling" along on the bottom, found beds of the round or hard-shell clam in such great quantities that there must have been thousands of bushels to the acre. This was in waters too deep to be recovered by the usual clammers' apparatus.

It is impossible to dredge for the soft-shell clam, as the shell is too delicate, and to pull them out of their bed would crush them. The abalone attaches itself to rocks, and it requires considerable force to break their hold, so there is no known means to recover them with surface ships.

The oyster industry is the only one that has thus far been developed by planting and cultivating methods, so that it is now a great industry, employing thousands of steam, internal-combustion and sail boats in their cultivation and collection for the market. The method employed by the largest growers is by the use of power boats, which drag "dredges," which are rakes with a meshed bag dragging on the bottom back of the tooth bar of the "dredge" to collect the oysters after they are "raked" or torn up from their beds. This is not a scientific method, for the reason that many of the oysters are killed by the heavy dredge being dragged over them. It is largely a "hit or miss" or "grab-in-the-dark" method, as it is impossible to "clean up" the ground in this manner. Some oyster grounds will produce from three to four thousand bushels of oysters to the acre. When "dredging" is started it is only necessary to drag the dredge a few feet before it is filled; then, as the oysters become thinner, the drag becomes longer. They drag in all directions across the grounds, but, as they cannot see the bottom, there are places they never hit, as the wind and currents prevent a systematic covering of the ground.

The design of submarine oyster-dredging vessel illustrated in Fig. 59 is such that the vessel goes down to the bottom direct, and the water is forced out of the center raking compartment so that the oysters may be seen by the operator in the control compartment. With only a few inches of water over them, headway is then given to the submarine and the oysters are automatically raked up, washed and delivered through pipes into the cargo-carrying chambers, as shown. Centrifugal pumps are constantly delivering water from the cargo compartments, which induces a flow of water through the pipes leading from the "rake pans" with sufficient velocity to carry up the oysters and deposit them into the cargo holds. In this manner the bottom may be seen, and by "tracking" back and forth over the bottom the ground may be "cleaned up" at one operation.

The design of vessel illustrated has a capacity of gather-

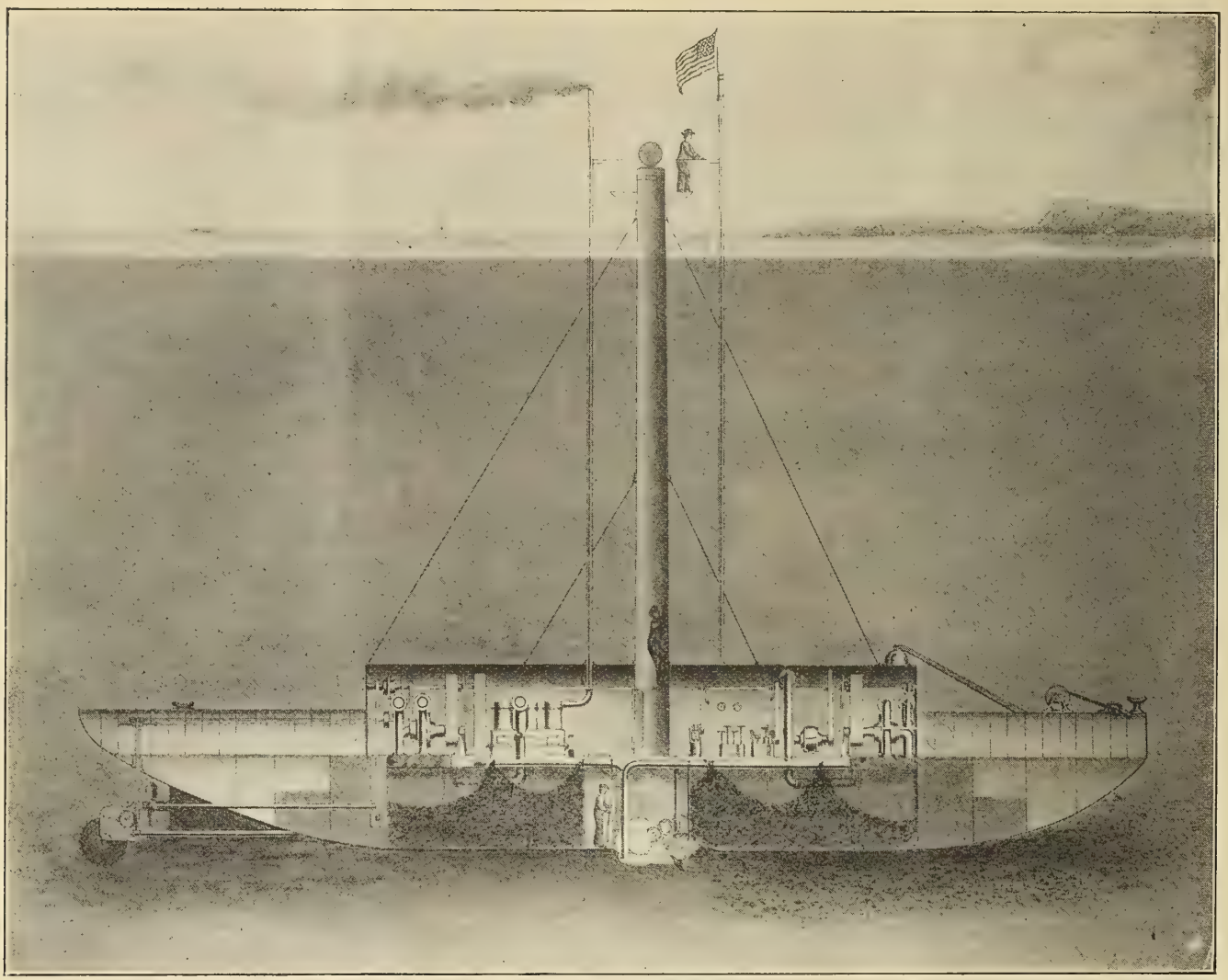


Fig. 59.—Submarine Oyster Gathering Vessel

By admitting water ballast into ballast tanks the vessel is allowed to sink to the bottom with sufficient weight to afford traction to the toothed driving wheels in the central operating compartment. This compartment is open at the bottom; water is prevented from entering it by the use of compressed air. As this compartment is well lighted the oysters may readily be seen lying on the bottom the full width and length of the compartment. When the boat is given headway the oysters are automatically transferred into the cargo holds as described on page 201

ing oysters from good ground at the rate of five thousand bushels per hour. The use of the submarine will make the recovery of oysters more nearly like the method of reaping a field of grain, where one "swathe" systematically joins on to another, and the whole field is "cleaned up" at one operation.

Submarines as Commerce Destroyers

Since beginning these articles ten months ago, nothing has as yet been invented or introduced to alter my opinion that the submarine is destined to be the premier weapon upon which maritime countries must depend for their defense. Much "ado" has been made about the "frightfulness" of the submarine. Nations that have used the submarine in an endeavor to prevent the transportation of munitions of war on the high seas—munitions of war or funds for the purchase of munitions of war which were intended to bring about the death of thousands of their own citizens—have, unfortunately, caused the death of non-combatants. They may or may not be justified in using the submarine in the manner in which it has been used, but the deaths of non-combatants have been trifling in comparison to the deaths of non-combatants that would have undoubtedly ensued had the submarine not existed as a factor in the war, and had the great fleets of surface ships been employed in the purpose for which they had been designed—in battling against one another, in the bombardment of coast cities, the chase and destruction of surface ships, etc.

Future submarines may be built that may avoid the apparent necessity of sinking without warning a hostile ship carrying contraband, but at the beginning of the war, and perhaps even to the present writing, *no submarine has had the ability to stop a high speed merchant ship except by sinking her during the few minutes* that she was in torpedo range. By means of the diagram shown herewith I hope to make this indisputable fact clear to every engineer or logical reasoner, irrespective of where his sympathies might lie, and I will describe the method that must be substantially pursued with most of the submarines now in commission.

In the diagram, Fig. 60, *A* is a merchant line ship that is pursuing her course in the direction of the arrow at a speed of, say, 20 knots; *B* is a submarine lying in wait for her enemies. If the submarine is lying in the light surface, or decks awash, condition, she might sight the approaching liner at a distance of ten miles. Her commander must immediately submerge to prevent discovery, otherwise the enemy ship would have time to sheer off and pass by out of range.

B then proceeds to maneuver for position so that *A* will

present a fair target of several hundred feet in length when *A* passes *B*. Now, *B* must not expose her presence until *A* gets within good torpedo range, which is not more than one-half a mile. For a swift-moving target, at a distance of one-half mile the periscope of a submarine is invisible to the naked eye of anyone on *A* when *B* is at rest, but as *B* can only fire her torpedoes in line with her own axis, she must secure and constantly maintain headway to direct her torpedoes towards the target *A*.

The navigator on *B* must also, by the use of his range-finders and direction indicators, be constantly employed in reading off distances and direction, and to also keep "tabs" on the speed of *A* to know how many degrees he must fire in advance of *A* to make a target. Now the

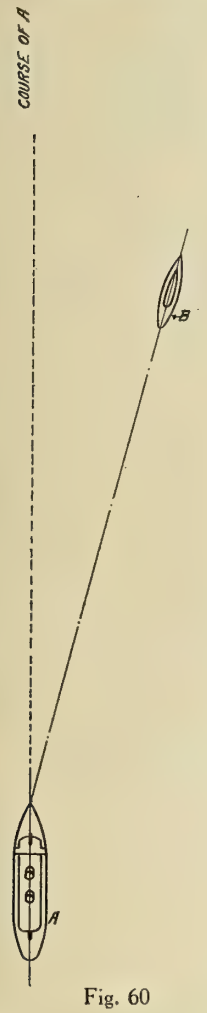


Fig. 60

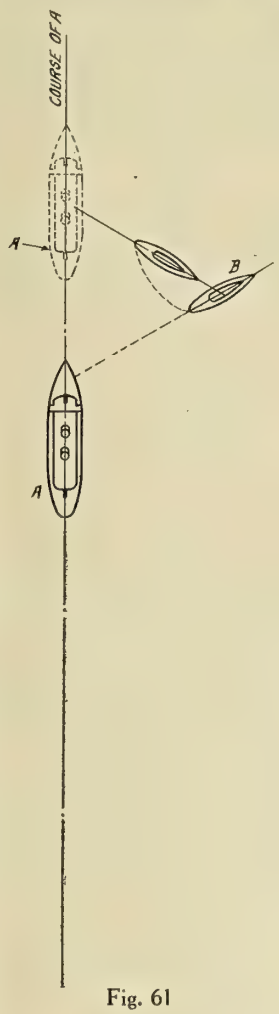


Fig. 61

speed of the torpedo is, say, 35 knots, and the speed of the ship *A* is 20 knots, the full length of the target, if of a ship 500 feet in length by 70 in breadth, is only presented when presenting a direct broadside to *B*. No submarine commander would be foolish enough to get in the direct course of *A*, as his target would be reduced to the thickness of the stem of the approaching vessel, as the water is deflected to either side of the stem of a moving ship, and the torpedo would probably be carried with the stream lines and not contact with the ship at all, or at best only effect a glancing blow. At the same time, the submarine commander would be placing his own vessel in jeopardy from being run down, so the maneuver is limited to the proceedings I have described.

When *A* has arrived in the position shown in full lines in Fig. 61 he must either cause *A* to stop before she reaches the position shown by the dotted lines, or his opportunity for capturing her is passed. Now it must be remembered that the speed of the ship is 20 knots and *B* is one-half mile from the line of the course of *A*. Therefore, *A* only has to run through one leg of a triangle which is also one-half mile, at her speed of 20 knots. This is at the rate of three minutes to the mile, or one and a half minutes to the half mile. Therefore, whatever is to be done must be done within one and one-half minutes. Now, when the periscope of *B* is first projected above the surface there is hardly one chance in a thousand that it would be even seen by the executive officer or lookouts on *A*.

The commander of *B* is now face to face with the following propositions, and he has only one and one-half

Fig. 62

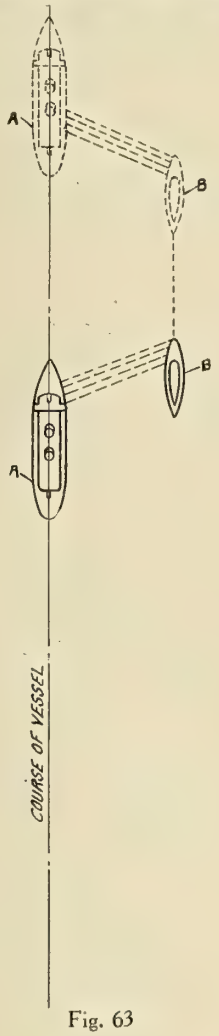


Fig. 63

minutes to decide it: Shall he let his country's enemy pass with perhaps sufficient contraband in the shape of munitions of war to slaughter hundreds or thousands of his own countrymen, or shall he quickly blow tanks, bring his own vessel up on the surface, blow whistles and set signals, in the endeavor to attract the attention of the executive officer of *A*, in the hope that the said executive will immediately give three bells and a jingle to stop—for, be it remembered, it takes considerable time to stop a large ship moving at 20 knots.

If the commander of the submarine adopts the latter course he puts himself and his own crew at the mercy of any marksmen who might be on *A*. Shall he do this? Or shall he stop the potential destructive energy that is being carried in the enemy's ship destined for the destruction of his own countrymen by using the only weapon

he has and fire a torpedo in the hope of stopping the enemy? I leave the answer to the patriotic sailors and soldiers of any nation.

I do not believe any submarine commander would, or has, wantonly sacrificed life, but what can he do? His difficulty lies in the limitations of the tools he employs. It is his bounden duty to his country to stop any ship carrying contraband. It was the recognition of this limitation of the present submarines that led me to reply to a telegram from the New York *Tribune* re the sinking of the *Lusitania* on May 11, 1915, in which I said: "I shall hope to see our government issue an official warning against Americans sailing on belligerent ships. I would also suggest that Americans sail only on American vessels which have been inspected by representatives of belligerents, certificates given that they do not contain contraband, and instructions cabled to their respective governments to give them a free passage."

The Congress of the United States recently voted, in substance, that it is beneath the "dignity" of a great nation to warn Americans to keep off from any belligerent vessel, whether carrying contraband or not, and that such vessels shall not be sunk and the lives of Americans thus imperiled. This is a very fine sentiment, and I hope it can be enforced from my American point of view, but from an engineering point of view that "dictum" interpreted means: No submarine shall be used to stop any belligerent vessels from carrying contraband on the high seas unless she first places herself at the mercy of the belligerent. Until all nations agree to this limitation of the submarine's function I shall hope to see Americans keep off belligerent vessels.

When this war is over and nations are again ready to "reason together," it is probable that new rules and regulations will be made governing the use of submarines, and it would seem logical that if a submarine got within torpedo range and could without question sink an enemy ship, and did not "for humanity's sake," that ship should no longer be used for the benefit of the enemy, but should be paroled in the custody of an officer from the submarine and proceed to a neutral port as a prize of war, not subject to recapture, and there interned until the final settlement of the differences. The ship would then be property which would be of value to both sides in the final settlement of accounts.

If the "dictum" is maintained that submarines shall not sink a belligerent merchant vessel without first giving warning and opportunity for escape of passengers and crew, that means that all submarines now in use by all nations are already obsolete, as far as their ability to stop armed merchantmen from carrying contraband of war is concerned. From the humanitarian point of view, it is unquestionably desirable that submarines have the ability to *capture* as well as *destroy*. Invention has already provided the means to accomplish this, but invention is much ahead of accomplishment.

Submarine of the Future

From a résumé of the foregoing, some predictions might be made as to the submarine of the future. *As such,* the submarine may be termed a perfected craft as far as general principles of construction and methods of control are concerned. Types are now in existence that can navigate successfully as surface boats and for hours at a time maintain within a few inches any desired depth between the surface and water bed. They may also remain at any desired depth while at rest, but not under way. They may also navigate over the water bed, and while on the bottom doors may be opened and the water kept from entering the vessel. Through these open doors divers may go out and conduct repairs to the submarine or work on or transfer cargo or stores from sunken wrecks or other submarines, mines may be planted or cables cut, communications may also be held between submarines while submerged, and so forth. But much yet remains to be accomplished regarding the submarine's auxiliaries, especially in relation to its propulsive equipment and machinery, both for surface and submerged work.

As previously stated, the internal combustion engine is ideal in principle for surface work, but at the present time it is complicated, noisy and unreliable. The storage battery and motor is in a more satisfactory stage for submerged work, in that the motor is almost noiseless and the battery will hold its energy without loss for considerable periods of time. There is room for improvement, however, in increasing the energy carried, both as regards weight and volume, if this can be done without a sacrifice of reliability. If future submarine commanders are to meet humanitarian requirements they must be in a position to demand "surrender" without placing themselves in a disadvantageous position. This calls for submarines of speed nearly approaching that of the surface vessel she wishes to capture. The writer had this in mind in the design of the Russian *Kaiman* and *Alligator* class of submarines, which were designed in 1905. Recently a notice appeared in the press that the *Alligator* had captured and brought into port a German armed vessel.

The United States submarine *Seal,* now called the *G* 1, as originally equipped, is the only vessel ever built in the United States that has had the capacity in principle to force the surrender of an armed surface vessel with speed superior to herself. Reference to the diagrams and Figs. 62 and 63 will, I hope, make clear the great advantage of vessels of the *Seal* type to capture rather than to destroy. The *Seal,* as originally built, was fitted with four torpedo tubes that could be trained to fire torpedoes to either broadside, and in her acceptance trials she successfully fired torpedoes in this manner, both submerged and on the surface. A photograph of the *Seal* firing one of her torpedoes from a broadside training tube is shown on page 560 of the December issue of this magazine.

Referring now to Fig. 62, the same tactics are pursued as described in relation to Fig. 60, except that the submarine, instead of heading *towards* the approaching enemy, heads *away* from her and remains at rest entirely submerged until the enemy reaches the same relative position as described in relation to Fig. 61. She then trains her broadside tubes on the enemy (see Fig. 63), blows out sufficient of her water ballast to enable her to start her engines under full power, breaks out her signals to surrender and starts at full speed ahead, *parallel* to the enemy ship, with four torpedoes constantly aimed at the vitals of the enemy. In Western parlance, she has "got the drop on the other fellow."

In this condition, if the submarine has a surface speed equal to the enemy's speed, she can keep a parallel course and maintain "the drop" as long as the commander of the submarine elects, or she may thus be escorted to and forced to enter a neutral port and there interned, or if the exigencies of militarism require it, she may be held until the passengers and crew have escaped and then be sunk.

Assume that the surface vessel had a speed of 20 knots, as before, and the submarine a speed of only 15 knots, which is now within the possibilities in a submarine of about 750 tons displacement, which would be of sufficient size to carry satisfactory broadside firing torpedo tubes. In this case the surface vessel would only have an advan-

tage of five knots in speed, which is at the rate of twelve minutes to the mile. Therefore, starting from the same relative positions, in which the submarine is one-quarter of a mile ahead at the start, it would take the 20-knot ship nine minutes to get one-half mile ahead, out of sure shot range, instead of only one and one-half minutes, as in the previous case. This would be ample time to enable the executive officer of the enemy ship to come to his senses and decide as to whether he preferred to surrender or be sunk. With four Whitehead torpedoes aimed at his ship, all of which could be instantly fired on the first sign of resistance, I think there would be only one thing for any sensible commander to do, and that is to surrender without demand of any kind whatsoever.

Certain other improvements might be mentioned and other tactics described, but owing to the present somewhat strained relations existing between the United States and other countries, I do not feel that any further disclosures should be made at the present time. For the same reason I feel compelled to refrain from describing what may be done with surface ships to make them unsinkable from the explosion of a Whitehead torpedo.

Appeal for Research Laboratory

In conclusion, I wish to make an appeal for the establishment of such a research bureau and laboratory as has been recommended by the civilian advisory board of national defense which has been created by Secretary Daniels.

The inventive genius of the country has been greatly handicapped by lack of capital and direction. It is a notorious and not very creditable fact that many of the country's inventors have been driven abroad for that recognition which should have been granted them in their own country. Invention and design, while similar in some respects, are quite distinctive in other respects. Invention requires original thought and freedom from set, formulated rules that have been designed to meet old and existing conditions, while design requires a knowledge of and studious analysis of all old things and its application to new. History shows that most of the great inventions that have made Americans prominent in the line of original thought have been made by men outside of the professions which benefited by them—Fulton, Ericcson, Hotchkiss, Sir Hiram Maxim, the Wrights and many others who have made important contributions to the art of defense, were men originally engaged outside the profession which ultimately benefited by their original conception.

To carry an invention to a successful conclusion requires: first, the original conception; second, capital; third, that which only capital can provide, analysis and the working up of the original conception into a practical machine or process. Undoubtedly, we have among our people many original conceptions that may prove of benefit to the nation for purposes of defense or offense. To take advantage of these, Congress should provide the capital. Army and navy officers should pass upon their utility, and could greatly assist in the development of such ideas as might be considered of importance by the Advisory Board. A thorough investigation should be conducted as to their originality and utility, to show that the ideas are worthy of development.

In carrying out this idea, it is obvious that the men comprising the Advisory Board are mostly men of large affairs and therefore they cannot be expected to give up their valuable time to detail work of making researches and investigations or to the supervision of design and experiments. A large force will, therefore, be needed to conduct such work and the amount of $1,500,000 (£308,000) now asked for will not go far if many lines of research and development are to be undertaken. Many private corporations have expended greater sums than this in the development of devices applicable to defensive purposes.

Provision should also be made to protect and properly compensate inventors for their ideas in proportion to the value of their inventions. It is only by treating them fairly that their confidence and co-operation may be secured.

It is to be hoped that private interests will not oppose the granting of money which the Advisory Board requests. It sometimes happens that private capitalists will do everything within the law, and sometimes things without the law, to protect devices of their own manufacture, thus to maintain a monopoly on their own goods. They have been known to do this, even where the safety of the nation is concerned.

(*The End.*)

Dry Steam in Scotch Boilers—Remarkable Performance of the Steamship Bristol

The steamship *Bristol,* built by the New York Shipbuilding Company, Camden, N. J., for the Coastwise Transportation Company, Boston, Mass., has just completed her maiden voyage from Norfolk, Va., to Boston, and as this vessel is fitted with Scotch boilers equipped with the first merchant ship installation of an invention for drying and superheating the steam *wholly within the boilers,* the very satisfactory performance of this vessel is of exceptional interest. The apparatus here used is the Lovekin internal superheater and drier, which gave throughout the trip nearly absolutely dry steam. As results are what count and as these results are quite phenomenal, we append selected data from the official report of the trial bearing on the steam quality and general features most important for obtaining a fair comprehension of the conditions.

One remarkable showing was that a drop of only 6 pounds was sustained from the boiler pressure to the admission line of the high-pressure cards, and of course the maintenance of steam by the engine containing only .554 percent moisture is something heretofore unheard of in ordinary Scotch boiler performance.

The dryness of the steam was ascertained by a Carpenter calorimeter used with great care, and the average quality of 99.44 percent dryness is remarkable when it is considered that the temperature of the gases at the inlet to the drier was never more than 500 degrees F. This particular installation was designed for dry steam only and not for superheat.

Preliminary Report S. S. Bristol for 80½-Knot Run on Maiden Voyage Between Norfolk, Va., and Boston, Mass.

Size of engine cylinders	23¼, 38½ and 67 inches diameter by 45 inches stroke.
Number of boilers	2
Type	Scotch
Total steaming time	53 hours 59 minutes
Speed, knots	10.78
Average revolutions per minute	75
Average indicated horsepower	1,795
Draft, leaving Norfolk, mean	24 feet 9 inches
Displacement, leaving Norfolk	9,630 tons
Vacuum, inches	27

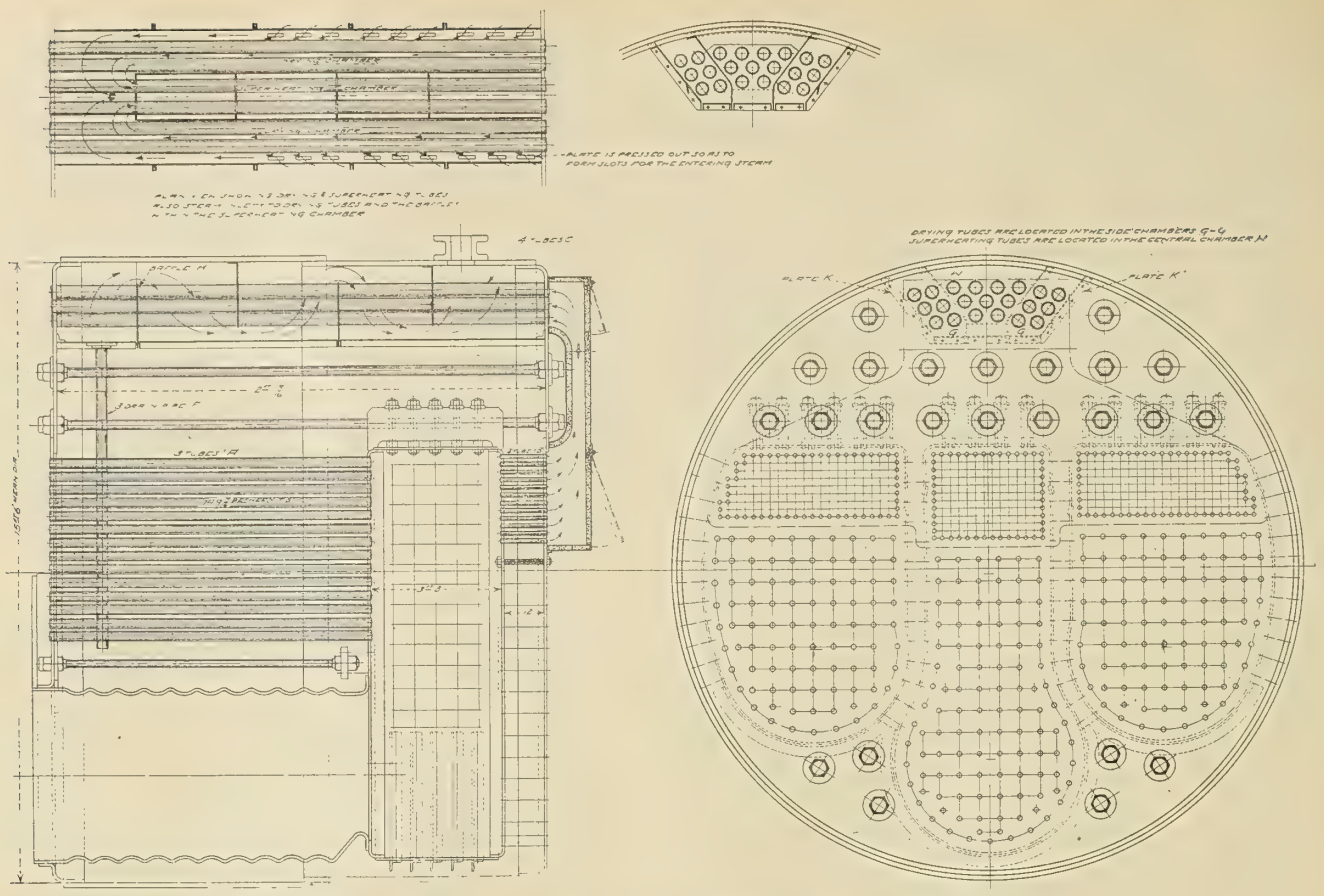


Fig. 1.—Scotch Boiler for S. S. *Bristol*, Fitted with Lovekin Internal Superheater and Dryer

Air pressure at fan, inches	1.3
Port boiler pressure, gage	161
Starboard boiler pressure, gage	163
Calorimeter temperature, port	297
Calorimeter temperature, starboard	300
Temperature of gases, inlet to steam heater	419
Temperature of gases, base of smokestack	496
Quality of steam, average both boilers	0.9944

The boilers after steaming 57 hours and standing with banked fires for 22 hours, upon opening of the superheaters and combustion spaces showed almost no signs of soot and no tube leakage, especially the superheater tubes. The boilers were equipped with Diamond soot blowers, and also Eckliff water circulators.

In the Lovekin steam drier and superheater fitted to the boilers of the *Bristol*, as shown in Fig. 1, steam enters through various lattice openings in the sides of the outer casings and passes over the drying tubes, where the greater part of the moisture is taken out of the steam. Any water remaining settles in the chambers *G G* and is drained to the lower part of the boiler by pipes *F*. The steam travels over the drying tubes towards the front of the boiler, where it enters the superheating chamber over baffle *H*. The steam is then compelled to pass in counter currents over the superheating tubes several times, due to the arrangement of the baffles as shown, and finally leaves the boiler at a point where gases of the highest temperature are entering.

It will be noticed that with this arrangement there can be no possibility of any moisture which may be carried along with the steam current affecting the final superheating of the steam, as the moisture is either all absorbed by the drying tubes before it enters the final superheating chamber, or is collected and drained off. All plates are

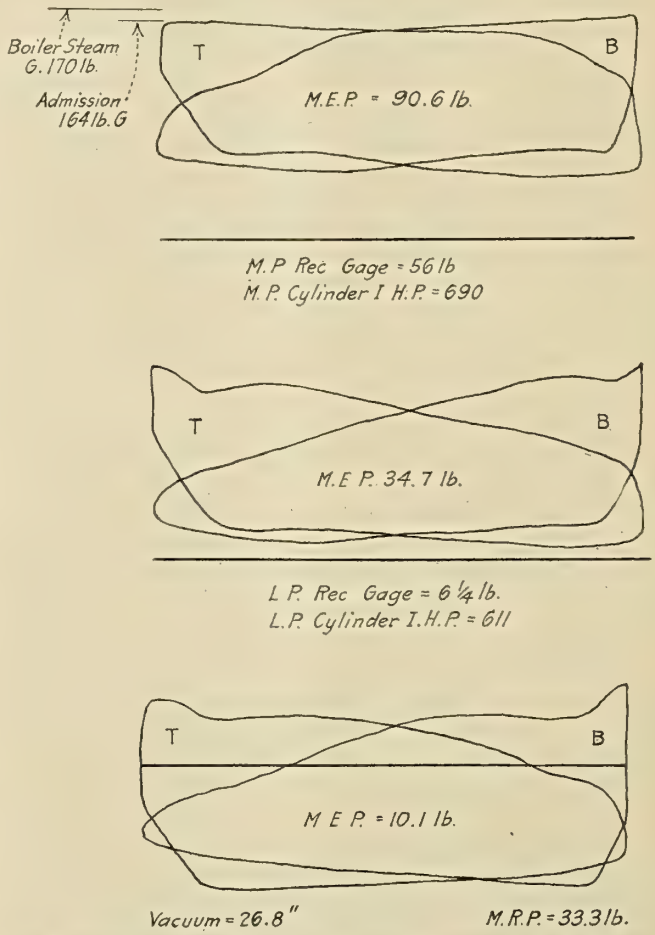


Fig. 2.—Indicator Cards Taken on Maiden Voyage of S. S. *Bristol*

made removable in order to give access to the boiler for examination.

The location of the slots for the entering steam is in a most favorable position for avoiding any water being thrown into same, which may be caused by rolling of the ship, or otherwise.

Heating Surface

Heat surface, tubes *A*	1,988 square feet
Heating surface, tubes *B*	178 square feet
Heating surface, furnaces	307 square feet
Heating surface, combustion chambers	312 square feet
Heating surface, back tube plates	84 square feet
Total heating surface *	2,869 square feet
Superheating surface, tubes *C*	266 square feet
Total heating and superheating surface	3,135 square feet
Volume, combustion chambers	325 cubic feet
Volume, furnaces	292 cubic feet
Total volume, combustion space	617 cubic feet
Calorimeter, tubes *A*	12.6 square feet
Calorimeter, tubes *B*	3 square feet
Calorimeter, tubes *C*	1.4 square feet

* Note: The heating surfaces are figured for oil fuel.

Efficient Steering

BY CHARLES R. PRATT *

The quicker a rudder moves the fewer degrees it has to swing and less retardation is given to the speed of the boat.

While this has been well known ever since the days of the Roman galleys, there has never yet been built a steering gear adapted to the actual work required in swinging a rudder.

Why keep hand wheels or power units constantly geared to put the rudder to "hard over" when it is seldom put there during an entire voyage? It only requires one-tenth the torque to put the rudder to a maximum cruising angle, and one-twentieth to an ordinary cruising angle; therefore, while it remains geared for "hard over," it only moves at one-tenth or one-twentieth of the speed that the power unit or hand wheels could move it during the whole voyage.

There is but one way to obtain "efficient steering" and that is by varying the leverage (gear) between the rudder and the power unit or hand wheels automatically by and in proportion to the sea pressure on the rudder. The writer has patented a device, shown in the illustrations,

* Consulting Engineer, 55 Liberty street, New York.

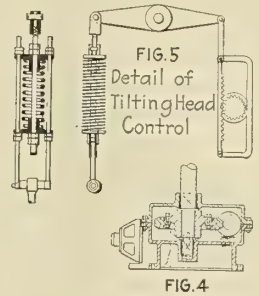


FIG.4
Detail of Pilot Motor Gearing

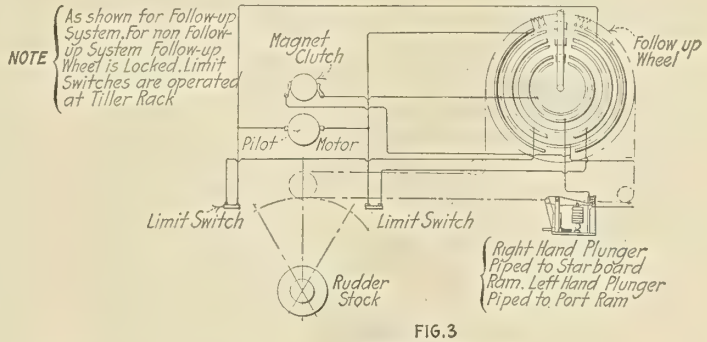

NOTE
As shown for Follow-up System. For non Follow-up System Follow-up Wheel is Locked. Limit Switches are operated at Tiller Rack
Magnet Clutch
Follow up Wheel
Pilot
Motor
Limit Switch
Limit Switch
Rudder Stock
Right Hand Plunger Piped to Starboard Ram. Left Hand Plunger Piped to Port Ram

FIG.3
Wiring Diagram

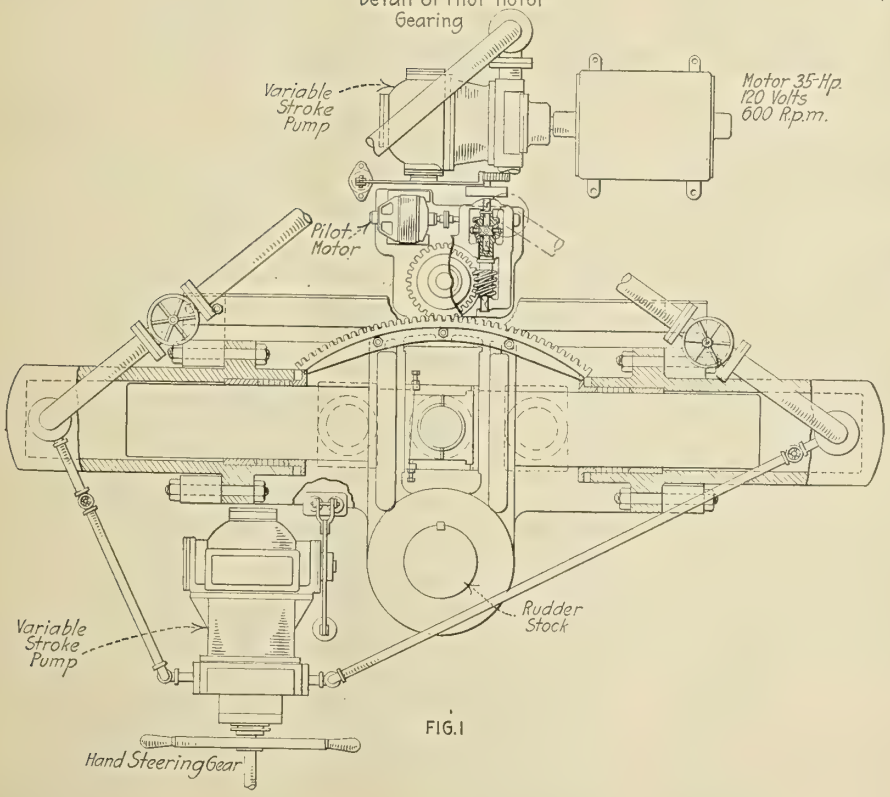


FIG.1

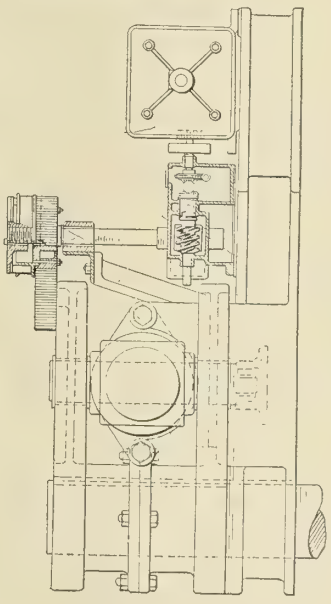


FIG.2

Details of Improved Steering Gear

which provides a very simple and efficient way to accomplish the required result. As hydraulic rams operated by a variable stroke pump have an average efficiency of 75 percent as against 25 percent for nuts and screws, or low pitch worm gears, this device, when using a 10 to 1 variation of leverage gives thirty times the rudder speed of the usual steering gear when using the same size power units, or, with the same number of men at the hand steering wheels, thereby making hand steering as efficient as power steering, and without having to slow the boat down to one-fourth speed when hand steering has to be used.

Hand Steering

Considering this device first for hand steering, we find by reference to Fig. 1 the tiller moved by hydraulic rams operated by a variable stroke pump, one pump driven by an electric motor for power steering and the other pump shaft turned by hand-steering wheels, closing the valves to one pump and opening them to the other pump shifts from power to hand steering. The stroke of the hand-steering pump is controlled by a lever which at one end has a spring keeping the pump at full stroke, while the rudder swings towards amidships and up to an ordinary cruising angle, giving the rudder thirty times the speed obtained by nut and screw gear, with the same number of men at the wheels.

When the action of the sea, the motion of the boat or swinging the rudder to a greater angle increases the sea pressure on the rudder, the fluid (oil) pressure on the ram (starboard or port) resisting same, operates one of the two small plungers on the other end of the pump control lever, overbalances the spring and shortens the stroke of the pump, thereby automatically increasing the leverage between the hand-steering wheels and the rudder in exact proportion to the increase of sea pressure on the rudder. By this device it requires no more manual effort to put the rudder to "hard over" than it takes to put it to an ordinary cruising angle. A quick adjustment of the spring pressure alters the automatic change of leverage to suit the number of men available for steering, whereby one man can do the work of twenty men at one-twentieth of the rudder speed.

Power Steering

In power steering instant precision of control between the steering wheel and rudder is quite as important as high rudder speed. With a telemotor the rudder lags several degrees behind the steering wheel in starting and runs several degrees beyond it in stopping, all of which has to be anticipated and controlled by a very expert helmsman. The usual electric steering gear has the unavoidable complication of magnet switches and resistance required to start and stop electric motors of over 20 horsepower, and the delayed action, due to the inertia of starting and stopping a heavy high-speed armature, thereby giving little, if any, better control than a telemotor.

The device illustrated uses a pilot motor operating on an entirely new principle which may be described as the combination of a member to be operated (the rudder), actuating means therefor (the pump), a pilot motor controlling said means, means dependent on the *torque* of the pilot motor for governing the actuating means, and means for governing the *torque* of the pilot motor. Heretofore pilot motors have controlled power units either by a step by step action or by varying their speed by field resistance, neither way giving results of precision. Reference to Fig. 1 shows the pilot motor geared to the tiller segment through a set of differential gears, the planetary pinions of which are geared to the lever controlling the stroke of the pump.

This pump shaft is driven continuously in one direction by the power unit (electric motor) and operates the rudder by reversing its stroke, thereby reversing the flow of fluid pumped to one ram and discharged by the other ram back to the pump. Its stroke control lever shown in Fig. 5 has a centering spring which keeps the pump at zero stroke when the rudder is stationary and opposes the torque of the pilot to increase the stroke in either direction (starboard or port) by action of a rack and pinion on the other end of the control lever. Incidentally, the full torque of this pilot motor is transmitted to the rudder, assisting the rams by that small amount, one horsepower.

As a rudder should always move at maximum possible speed, this pilot motor is started instantly at full speed in either direction with no armature resistance. This overcomes the centering spring, starts the pump off zero stroke and starts the rudder in about one-tenth of one second after closing its circuit with the steering wheel, getting the pump to full stroke and the rudder to full speed in the least time possible. In stopping the rudder, a magnet clutch breaks the connection between the pilot motor gears and the rack and pinion on the pump control lever, allowing the centering spring to snap the pump back to zero stroke and stop the rudder instantly.

The precision of this control as applied to a variable stroke pump operating an elevator in the testing tower of the Otis Elevator Company indicates a rudder control of one-twentieth of one degree with the steering wheel, as against several degrees by any other type of steering gear.

Fig. 3 shows the electric control, which can be either follow-up or non-follow-up, and also shows a rheostat operated by two spring-balanced plungers piped to each rudder ram, whereby the sea pressure on the rudder varies the torque of the pilot motor by its armature resistance and thereby the stroke of the pump and the leverage between the power unit and the rudder the same as in hand steering, whereby a one horsepower motor can handle the rudder as well as a 30 horsepower motor operating nuts and screws, or the same size motor can move the rudder at ordinary cruising angles at thirty times the usual speed, or any intermediate condition that may be required can be obtained.

The rudder is held stationary against leakage of rams and pump by a friction clutch shown in Fig. 2 and by the locking action of the worm gear on the segment pinion shaft. This avoids the continual action of the pump at short stroke to hold the rudder stationary, but it slips against excessive sea pressure when the relief valves open and allow the rudder to swing towards amidships. With a follow-up control the rudder is automatically returned instantly to its exact original position.

The novelty of this device and the fact that these drawings show every working part suggests a relatively high cost and complication, whereas a careful investigation will show a reduction in cost in comparison with other steering gears, with simplicity in construction and reliability in action. Its reliability has been proved beyond all question by the "Hele-Shaw-Martineau steering gear," which has every element of this device except the *automatic* change of leverage by the sea pressure on the rudder and that it controls the pump by telemotor instead of by this electric pilot motor. Up to January, 1914, there were 35 boats of from 600 to 28,000 tons equipped with this steering gear. Any boat now using a Hele-Shaw-Martineau steering gear could have every element of this control applied to it quickly and at small expense.

The Hele-Shaw pumps and the pumps shown on this drawing are piston or plunger pumps, pumping oil at 1,000 pounds pressure per square inch, without valves or packing, and with an efficiency of 90 percent.

Letters from Marine Engineers

Discussion of the Design and Handling of Marine Engines, Boilers and Auxiliaries—Breakdowns at Sea and Repairs

This department is open to all readers of the magazine for the discussion of affairs in the engine room. All letters published are paid for at regular rates. Your ideas or experiences will be mutually helpful and interesting to other engineers. Write your letter now.

A Handy Kink

Readers of MARINE ENGINEERING may be interested to learn that a large percentage of the burned-out Mazda or Tungsten lamps may be easily repaired. It is a fact that if the broken ends of the filament can be brought into contact with the curent on the lamp, the ends will fuse together and become permanently welded. If the filament is broken in only one place the ends may be again brought into contact by screwing the lamp into a portable socket and, holding it in various positions, gently tapping the lamp with your finger or gently shaking it to set up a little vibration in the filament. If the ends of the broken filament can be made to touch, the current flashes across and the job is done and the lamp is practically as good as ever. A little practice enables one to become quite adept in applying this kink to practical economy.

B. SAWYER,
Chief Engineer, Steamship *Northland*,
Norfolk & Washington Steamboat Company.
Norfolk, Va.

How Feed Water Should be Discharged Into a Scotch Boiler

There are several fundamental principles which govern the location of the feed outlet in a Scotch boiler, and the designer must bear in mind the conditions under which the boiler is to be used.

1. As a general rule, feed-water heaters are installed in all first-class plants, either stationary or marine, and the feed water enters the boiler at about 200 degrees F., and the customary practice is to discharge the water above the tubes just below the water level, not from the open end of the pipe, but through two or more branches led over the tubes with the ends closed and the sides perforated with small holes, the combined area to be 1½ times the area of the feed pipe; in this manner the water will be uniformly sprayed over the surface.

When water enters the boiler at 200 degrees F. in the manner described it has its best advantage and is rapidly evaporated into steam. Due to this high temperature, it has an advantage of 2 pounds per horsepower per hour.

If this discharge were led to the bottom of the boiler, where the water is fairly cold (and it is seldom any other way unless a system of perfect circulation is provided) the effect of the heater would be entirely destroyed.

To bear me out in this I will cite a recent instance of a small tug boat with a compound engine non-condensing one Scotch boiler with 1,200 square feet heating surface and 40 square feet of grate surface and no heater. At the suggestion of the owner, a good feed-water heater was installed. After several weeks there was no noticeable difference in the consumption of fuel, and it was then that the writer discovered that the internal feed pipe discharged its water near the bottom of the boiler. This was immediately changed and placed above the tubes in the manner previously described, and better results were noticeable from the start. The exact extent of economy was not recorded, but was about 140 pounds of coal per hour, enough to prove that the change was very desirable.

2. If there is no feed-water heater installed and the feed is made up with cold water from the fresh-water tanks, it is the writer's opinion that the proper place for the discharge is near the bottom of the boiler, using a tee fitting on the internal pipe with an open end branch in each direction. This, of course, should not be so close to the bottom as to stir up sediment.

When water enters the boiler thoroughly cold it is advantageous to have the discharge as far away from the water level as possible. Cold water entered at the water level prevents proper evaporation.

3. The United States Steamboat Inspection rules require all boilers to have two feed connections, main and auxiliary; sometimes the auxiliary feed is connected only with the injectors and is seldom used except for supplying the boilers sufficiently to keep auxiliary machinery running when the main units are shut down; the discharge then should be placed over the furnaces and sprayed over the surface, as in case No. 1.

When the auxiliary feed is supplied with an independent feed pump, using cold water, the same case applies, and the discharge should be placed over the furnaces, this being the most effective part of the boiler when the fires are banked.

4. In some plants the auxiliary feed pump is used in case of an emergency, and is connected to the main feed line and heater, and should discharge over the tubes just below the water level, as in case No. 1.

5. The feed valves can be located in any convenient place on the boiler; shell, backhead or front head, if sufficient space can be found, but in no case should the discharge terminate above the water level in the steam space, for the reason that it produces excessive priming and also excessive air hammer in the feed line. Great quantities of air pass into the boiler with the water at all times, which produces a certain amount of hammer in the line. To overcome this an air chamber of ample capacity should be placed on the discharge side of the feed pump or in some convenient place in the feed line.

New York.

R. R. ROANE,
Mechanical Engineer.

Analyzing a Marine Steam Plant

To the practical marine engineer it often remains a mystery why one plant is more economical than another, and why one engine differs from another, although built from the same drawings. Many boats are known to have something wrong with them, yet there seems no way of getting at the trouble without turning the ship over to experts, which is usually expensive, inasmuch as it either lays up the ship or puts the company to the expense of carrying several experts for a trip or two, besides their remuneration. To that growing class of marine engineers who are willing to think, it is not a difficult task thoroughly to check up the possibilities of their plants; nor does one have to be highly educated to do it. Before taking up the steps of an analysis, it is well to review briefly the governing features of design.

A small, medium speed boat operating on short trips need not go to the expense of putting in a highly economical plant, and compound engines will often answer.

Higher speed boats, or those intended for longer trips, are constructed with more economical plants. The higher the steam pressure, the greater the economy and the number of expansions. The higher the steam pressure, the more expensive is the engine and the greater the amount of coal burned per square foot of grate surface, although this is compensated for by greater efficiency.

The number of expansions of steam is the volume of steam at cutoff in the high-pressure cylinder, divided into the total volume of steam in the low-pressure cylinder, just before exhaust. These expansions are divided into two, three or four stages, known as compound, triple, etc.

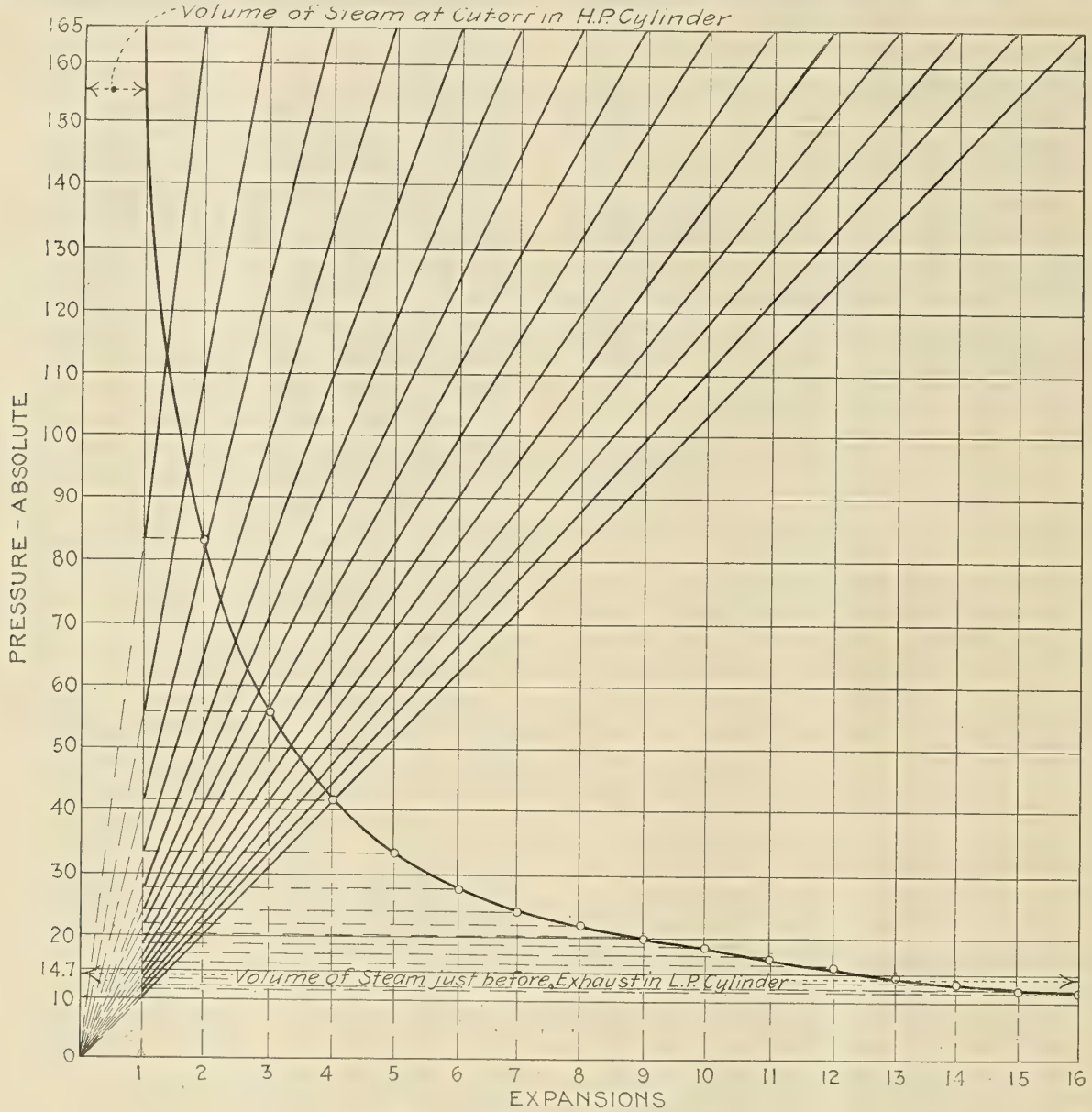


Fig. 1.—Theoretical Expansion Curve

The clearest way of showing the possibilities of any marine plant is by means of the combined card, which shows the possibilities of the entire plant, from boilers to condenser. Suppose that we have a regular triple expansion plant of average design. Such a plant will have about sixteen expansions, when the high-pressure cylinder is cutting off at three-quarters of the stroke. Assume that this plant carries 150 pounds of steam, say about 165 pounds absolute pressure, reckoned from vacuum. Tables of expansion of steam, which can be found in almost any engineers' pocket book, tell us that for sixteen expansions the mean pressure will be 23.5 percent of the absolute boiler pressure and cutoff at 6.2 percent; that is to say, that if the boiler pressure were put in the low-pressure cylinder and the cutoff placed at 6.2 percent (which, of course, is impractical), the low-pressure cylinder would generate the full power of the engine, with the same number of expansions. This gives an easy method of figuring what horsepower an engine ought to develop, but it would not be economical in practice.

The reason that more than one cylinder is used is to enable the higher pressure cylinders to stay hot, without being cooled down by the low-pressure steam. The same table tells us that at sixteen expansions the terminal pressure would be 6 percent of the absolute boiler pressure. This would be six times 165 divided by 100, or 9.9 pounds per square inch. This is equivalent to a vacuum of about 10 inches of mercury. In practice, the vacuum is always lower than the theoretical line.

Let us take now a piece of drawing paper and near the bottom draw a zero line. At right angles to this draw another zero line. Divide the vertical line into divisions representing 165 pounds absolute pressure. Divide the horizontal line into 16 expansions. Draw in the one vol-

ume of steam and the 9.9 pounds terminal pressure. Then by means of the dotted lines, as shown, draw in the theoretical expansion curve, known as the hyperbola. It is drawn through the corners of the squares, as shown in Fig. 1.

We now have as a starting point a triple expansion plant which expands the steam sixteen times, beginning with 165 pounds absolute pressure, and with ten inches of mercury in the condenser. We have the theoretical expansion line and the theoretical mean effective pressure for the whole engine, referred to the low-pressure cylinder, thus showing the theoretical horsepower for any engine. We will begin at the boiler and go through an entire average plant, noting how closely it complies with the theoretical results, and why it falls below them.

Fuel

A British thermal unit is the amount of heat necessary to raise the temperature of 1 pound of water 1 degree F., and 2,545 B. T. U. is one thermal horsepower—that is to say, if 1 pound fell 33,000 feet in a minute, or was projected out of a cannon at a velocity of 33,000 feet per minute and struck a target, 2,545 B. T. U. would be given off in heat, etc. This is what causes heat and sparks on impact, and is the reason some blacksmiths can hammer a piece of iron until it is hot.

Good coal has a calorific capacity of 12,500 B. T. U. per pound, poor coal a calorific capacity of 8,500 B. T. U. per pound. As a rule, it is poor economy to buy cheap coal, because it has 25 to 35 percent less heat and costs only 5 percent to 10 percent less. Pure anthracite contains about 14,000 B. T. U. per pound, which accounts for its excellence as a steam coal. Fuel oil averages 19,000 B. T. U. per pound. It occupies about the same volume, ton for ton, as coal. The light oils have greater calorific value per pound, but less per barrel, and they are usually paid for by the barrel. With the heavy oils the converse is true.

The average triple expansion marine engine working on sixteen expansions will use a little less than two pounds of good coal per horsepower hour, or a little more than two pounds of poor coal. The same engine will use, roughly, a pound and a quarter of fuel oil per horsepower hour. For the remainder of this article, coal will be considered exclusively, for the analysis of a marine plant.

The first possible source of loss is in firing. When the fires are covered with a fresh layer of coal, a small portion of the live hot coals should be left uncovered, so that the heat can ignite the immense amount of tar and gas that is distilled from the fresh coal, and otherwise appears as green and black smoke in the stacks. As soon as the furnace door is closed after firing, the door shutters should be opened to furnish air to burn the vapors and tars referred to above. When the coal becomes thoroughly incandescent, the shutters should be closed, since there is no longer an excess of combustible matter above the fire bed.

Another great source of loss is excess air. The amounts of air admitted to the furnaces should be regulated so that the stacks do not smoke, but they should almost smoke. It is easy to run two or three times as much air as is necessary through the furnaces, and every 100 percent means an approximate loss of about 8 percent in fuel, since it costs just as much to heat air as to heat water, pound for pound.

There is only one way to know how good combustion is. That is to use a simple device known as an Ellis tube and test a sample of the smokestack gases for carbon dioxide. Any engineer can use the tube and any chemical house sells and explains how to use it. Perfect combustion produces about 16 percent carbon dioxide in the stack gases. If the instrument shows 14 percent carbon dioxide, combustion is very good. If the percentage drops to 6 percent, there is either too much air going through or not enough. Another chemical is put in the tube and another sample tested for carbon monoxide (partially burned carbon). If anything over 3 percent of this shows, it is fairly certain that there is not enough air under the grate bars. If the quantity of monoxide is only a trace, it is pretty certain that there is too much air, which is the more usual fault.

The efficiency of the combustion cannot be judged very accurately by the smoking of the stacks, because when fresh coal is laid on no amount of air will burn the smoke because there is nothing to ignite it. On the other hand, the stack may not be smoking at all, yet that hotbed of fuel will be pouring out quantities of carbon monoxide, which is invisible and all of which represents a fifty percent waste while it lasts. There are now on the market recording devices which automatically take and test samples of stack gases and trace a curve, which on a large ship is a very valuable index of how firing is being conducted, and shows at just what times of day the bad firing was done. These may be procured from several large chemical firms and cost from $200 to $300 (£41 to £62).

A good boiler, working under favorable conditions, will absorb 85 percent of the heat generated by the fuel, 75 percent is more usual, and 50 percent is not unknown, particularly when the boiler is in bad condition from scale or salt or soot.

Suppose that the feed water enters the boilers at a temperature of 120 degrees F., which would be an average condition, and must be heated to 165 pounds pressure. By reference to the tables of saturated steam, we find that the temperature of saturated steam at 165 pounds absolute pressure is 365.7 degrees F. In other words, water boils at 365.7 degrees F. at 165 pounds absolute pressure. To be heated from 120 degrees to 365.7 degrees would require 366 minus 120, or 146 B. T. U. for one pound of water.

Next, we find by the same steam tables that it takes 855 B. T. U. to evaporate the water into steam at 165 pounds absolute pressure. This gives a total of 146 plus 855, or 1,001 B. T. U., as the heat necessary to evaporate a pound of water, first heating it from 120 to 366 degrees F., then evaporating it into steam at 165 pounds absolute pressure. We will call it 1,000 B. T. U., as a more convenient number.

Assume that a pretty good coal, having a calorific value of 12,500 B. T. U., is used; then 12,500 B. T. U. divided by 1,000 gives 12.5 pounds as the amount of water each pound of coal should evaporate theoretically. If the boiler is in pretty good condition and receives 80 percent of the heat, the coal may be expected actually to evaporate 80 times 12.5 divided by 100, or 10 pounds of water, which accords with usual practice.

In order that different boilers may be compared, since they are not all operating at 165 pounds pressure, nor does the same boiler operate at the same pressure in different tests, it is customary to reduce the actual evaporation to a common standard, known as the equivalent evaporation from and at 212 degrees F. This is done by first dividing the actual heat received by the boilers by 965.7 B. T. U., which is the amount of heat necessary to evaporate into steam 1 pound of water at 212 degrees F. The B. T. U. actually received by the boiler per pound of fuel burned is obtained from a test in which all temperatures are obtained, also the quantities of feed water and the time.

The "boiler horsepower" is the evaporation of 30 pounds of water per hour from 100 degrees F. to steam at 70 pounds pressure. The heat required is 33,305 B. T. U., and is almost exactly thirteen times a theoretical horsepower of 2,545 B. T. U. The basis for this rating is that the ordinary simple slide valve engine uses about 30 pounds of water per horsepower hour. It might be mentioned here that a compound engine uses 13 to 16 pounds of water, a triple expansion engine 10 to 12 pounds, a quadruple expansion engine 8 to 11 pounds, and steam turbines all the way from 12 to 17 pounds per horsepower hour. Greater economy could be realized in marine turbine plants if much higher speeds could be provided for in the design.

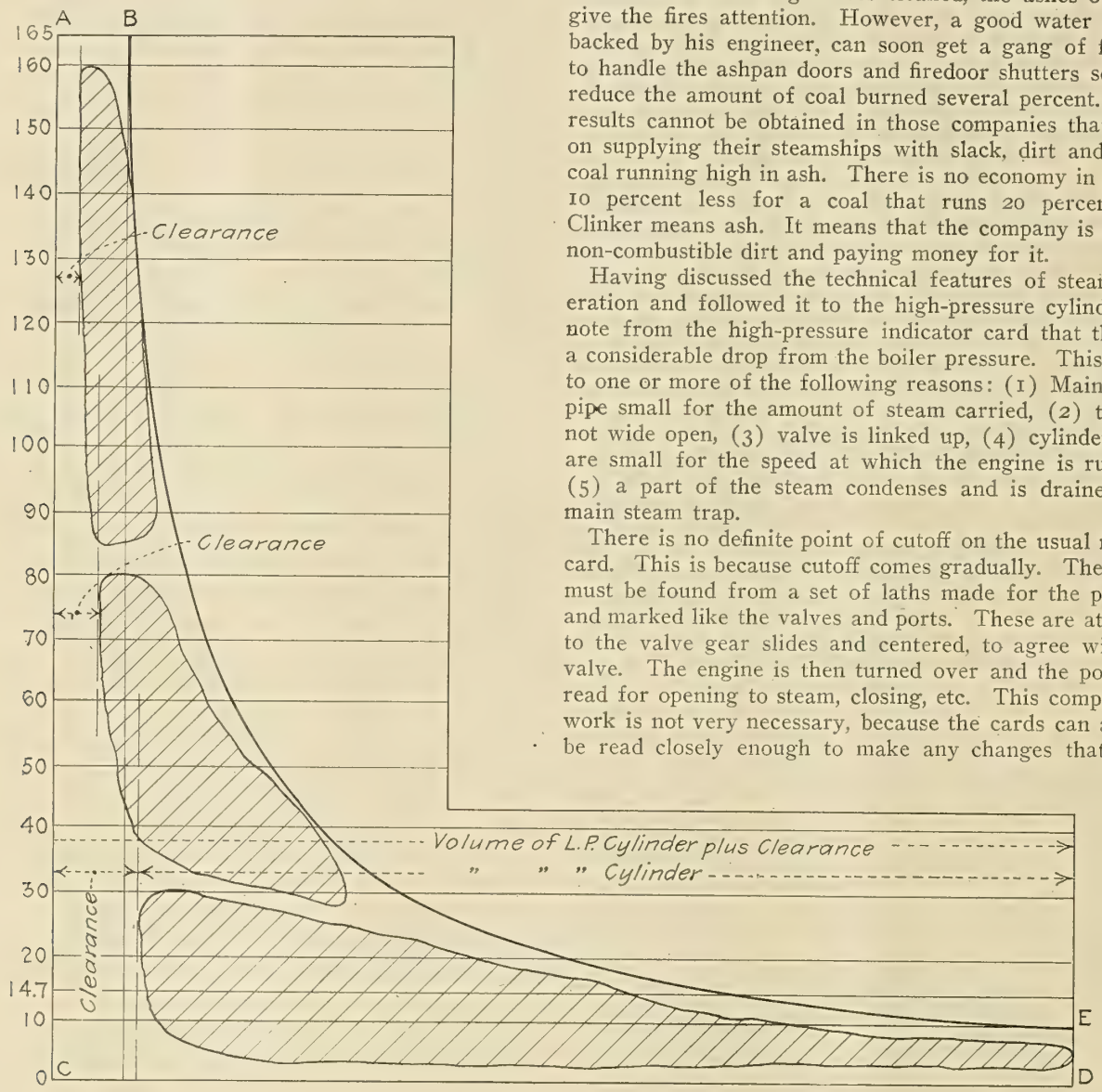


Fig. 2.—Combined Indicator Card

Scotch boilers generally evaporate from 7 to 12 pounds of water for each square foot of heating surface. They have from 5 to 9 square feet of heating surface per boiler horsepower. A boiler horsepower would, for a triple expansion engine, provide steam for an average of 2.5 horsepower, allowing 12 pounds of steam for each horsepower. The ratio of the heating surface to grate surface varies from 16 to 30. The stack temperature of the average Scotch boiler ranges from 700 to 1,000 degrees F. Water-tube boilers are lighter and more efficient than Scotch boilers, but much more expensive as a rule.

Scotch boilers burn from 20 to 40 pounds of coal per square foot of grate surface per hour. They represent the heaviest fires used in steam engineering, the fire bed being from 6 inches to 16 inches thick, and the closest attention is necessary to get efficient combustion. Unfortunately, there is so much hard work, dirt and heat in a ship's stokehold that there is little inspiration for efficient work other than to get fires cleaned, the ashes out, and give the fires attention. However, a good water tender, backed by his engineer, can soon get a gang of firemen to handle the ashpan doors and firedoor shutters so as to reduce the amount of coal burned several percent. Such results cannot be obtained in those companies that insist on supplying their steamships with slack, dirt and cheap coal running high in ash. There is no economy in paying 10 percent less for a coal that runs 20 percent ash. Clinker means ash. It means that the company is buying non-combustible dirt and paying money for it.

Having discussed the technical features of steam generation and followed it to the high-pressure cylinder, we note from the high-pressure indicator card that there is a considerable drop from the boiler pressure. This is due to one or more of the following reasons: (1) Main steam pipe small for the amount of steam carried, (2) throttle not wide open, (3) valve is linked up, (4) cylinder ports are small for the speed at which the engine is running, (5) a part of the steam condenses and is drained into main steam trap.

There is no definite point of cutoff on the usual marine card. This is because cutoff comes gradually. The cutoff must be found from a set of laths made for the purpose and marked like the valves and ports. These are attached to the valve gear slides and centered, to agree with the valve. The engine is then turned over and the positions read for opening to steam, closing, etc. This complicated work is not very necessary, because the cards can always be read closely enough to make any changes that seem necessary. If they cannot be seen, it is not likely that the fault is important.

It is very necessary, though, in order to analyze a steam plant, to know the volumetric clearance of each cylinder. This may be found by careful measurement and calculation of the clearance volume at each end of each cylinder, and the space between the cylinder and the valve in the port passage. Another way is to seal the piston rings with soap or grease and fill the clearance space with water, the amount being carefully measured as it is poured in. The clearance space should be determined in cubic inches. These clearance spaces are the starting point for the analysis of a steam engine plant.

Now taking Fig. 1, trace the curve and two zero lines

on another piece of paper, leaving out the construction lines. This is shown in Fig. 2. The terminal pressure line should be traced, also the initial volume line.

The first thing to do is to put in the atmospheric line, usually at 14.7 pounds pressure. Now the volume of steam, just before exhaust, is the volume of the low-pressure cylinder plus the volume of port, etc., filled with steam. Add these two together and make the total equal to the length of the botom line. Then lay off the clearance on the bottom line.

The clearance for a cylinder with a slide valve will run from 3 to 5 percent, and with a piston valve anything up to 30 percent of the cylinder volume. Divide the low-pressure cylinder volume line into ten parts and draw in the low-pressure indicator card as shown in Fig. 2. In the same way lay off the clearances of the intermediate and draw the cards, measuring everything from the atmospheric line and the clearance lines. The cards will look very different, because the scale one way will look different from the scale the other way. Their length is in terms of volume instead of stroke, and their height is in terms of absolute pressure instead of cylinder pressure. The five-cornered space *ABEDC* represents the theoretical power of steam between 165 pounds and vacuum, and the area of the cards represents the power obtained.

The losses due to clearances, condensation, drop in receivers and imperfect vacuum are clearly visible. If any of the valves were leaking it could be easily seen. If the high-pressure valve leaks, the high-pressure card would show a higher exhaust line. The intermediate-pressure and low-pressure diagrams would rise to the theoretical line, or even rise above it, which is an absolute indication of trouble. If the intermediate-pressure valve leaked, the intermediate-pressure card would fall away from the theoretical line and the low-pressure card would rise in proportion. Also, the exhaust line of the high-pressure would be lower. An exhaust pipe that is too small, or a condenser that is too small, raises the low-pressure exhaust line. Triple expansion engines are usually designed to divide the work evenly between the cylinders, and the cards should be approximately equal in the combined diagram, but no great harm is done if the cylinders develop different amounts of power.

In figuring the steam consumption of a triple engine, it is best to take a point on the high-pressure card where it touches the theoretical line and figure from this, by means of the steam tables, as being the most accurate way, especially as the clearance is here properly included. It is almost impossible to figure steam consumption from an ordinary indicator card unless placed in the combined diagram, because of the error due to clearance, and due, also, to the small scale of the average card. Study and reflection will show how practically any fault with the proportions or adjustment of a steam engine of the multi-stage type can be found on a carefully constructed combined diagram, considered in conjunction with other available data about the boat.

We note from this diagram an average exhaust pressure of 3 pounds absolute. The steam tables tell us that the latent heat at this pressure is 1,015 B. T. U. and the boiling point 141 degrees F., which agrees with practice. Therefore 1,015 B. T. U. will have to be wasted in condensing this steam to water again. Making allowances for slight differences of calculation, we note that we have the same amount of heat as we started with, 1,000 B. T. U., but that it is at a lower temperature and higher latent heat. The condenser removes this heat and condenses the steam, thus allowing the lower exhaust pressure to approach the theoretical. With the exception of a few losses the heat with which the steam started is still in it when it enters the condenser, but in the form of increased latent heat, since the temperature has dropped from 365 to 141 degrees F. The useful power, therefore, might be calculated as follows:

Heat from 1 pound of coal.....	12,500 B. T. U.
Pounds of steam from 1 pound of coal....	10 pounds
Fall of temperature (365 minus 141).....	124 degrees F.
B. T. U. realized in useful work.....	1,240 B. T. U.
Ratio of useful work to original heat.....	10 percent

The last result, as worked out, accords with practice, and is known as the thermal efficiency. Note that although 12,500 B. T. U. is put into 10 pounds of steam, only 1,240 B. T. U. ever comes out. The remainder is thrown into the condenser, as we can easily see, as follows: The latent heat is 1,015 B. T. U. times 10 pounds of water, or 10,150 B. T. U., to the condenser. This, plus 1,240 B. T. U., gives 11,390 B. T. U., which, with the losses from radiation, etc., and the fact that the water enters the condensers hotter than it enters the boilers, accounts for the original 12,500 B. T. U. in the pound of coal.

In examining combined cards, a defective air pump, or one that is too small, will affect the card much the same as a condenser that is too small, but this trouble can always be identified by giving consideration to the condenser area in connection with the amount of steam condensed. If the area is ample and the temperature is normal and the tubes clean, the trouble places itself by elimination on the air pump.

The steam used by auxiliaries can be arrived at very accurately by taking cards on the main engines when the auxiliaries are exhausting into the receivers, and making a combined diagram; then taking another set of cards, when the auxiliaries are exhausting into the atmosphere, and making another combined diagram, and then comparing the two. The exhaust of the auxiliaries very seriously affects the economy of the ship as a whole, and very few engines are properly proportioned in respect to taking care of the exhaust of the auxiliaries into their receivers under the actual operating conditions. Many ships have had large auxiliaries added, such as ice machines, electric lighting plants, etc., and others have had their valve gears changed with respect to the main engines only, no consideration being given to auxiliaries. The horsepower of auxiliaries may not be very great, collectively, but it should be remembered that their consumption of steam is very high. Steam pumps use 50 to 100 pounds per horsepower hour. Slide valve engines use 30 or more pounds, and some engineers regularly leave passover valves open, or partly open, if the ship is crowded for time, and the boilers can make the steam.

Other phases of analysis of marine plants, reduced to a practical basis for the "boys," will follow when our editor feels that he can make room for them. E. N. Percy.

Shipyard Improvements.—A wooden floating dry dock 450 feet long, with a capacity for handling vessels up to 7,500 tons, is being added to the Hunter's Point plant of the Union Iron Works, San Francisco, Cal. At Seattle, Wash., the Skinner & Eddy Shipbuilding Corporation is building a new shipyard double the size originally planned, in order to accommodate the additional business which the company has secured. At Aberdeen, Wash., the Aberdeen Shipbuilding Company, recently incorporated for $10,000 (£2,050), is planning the construction of a small shipbuilding plant. At the Mare Island Navy Yard, San Francisco, Cal., the improvements projected include the addition of a large floating crane.

Questions and Answers for Marine Engineers

Inquiries of General Interest Regarding Marine Engineering and Shipbuilding will be Answered in this Department

CONDUCTED BY H. A. EVERETT *

This department is maintained for the service of practical marine engineers, draftsmen and shipbuilders. All inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given permission to do so. Indicator cards taken from marine engines will be carefully analyzed, the defects pointed out, and the horsepower calculated, provided complete data are sent with the cards.

Best Method of Reducing Power

Q.—Will you please tell me which is the better thing to do when operating for considerable periods of time at half power to get the power reduction by throttling down or by linking in on the reverse shaft? The vessel is equipped with the ordinary Stephenson link gear. P. R.

A.—It is more economical to adjust the valve gear by linking in.

Schools of Naval Architecture and Marine Engineering

Q.—Will you please give me a list of the colleges in this country which give courses in naval engineering? STUDENT.

A.—The institutions of college grade in this country which give courses open to the public in marine engineering and naval architecture are the Massachusetts Institute of Technology and the University of Michigan. Webb's Academy, in New York, also gives courses, and many short, practical courses are given by the various branches of the Y. M. C. A. and shipyards.

Operation of Westinghouse Refrigerating Machine

Q.—Will you please give in your valuable magazine a brief outline of how this new Westinghouse steam jet refrigerating apparatus operates? I cannot see how it is possible by the use of steam to get any temperatures which are low enough to be of any use for refrigeration purposes. R. E.

A.—The underlying principle is that the vaporization of a liquid absorbs a large amount of heat, and the device is one to produce a rapid vaporization, which, being maintained as a continuous operation, ultimately produces temperatures as low as 30 degrees F. The steam jets are simply used as ejectors to produce the very high vacuum in the chamber, where vaporization takes place. The brine is drawn into this vacuum chamber, to which the steam nozzle ejectors are attached, is mechanically atomized, a part of it vaporizes, thereby cooling the remaining liquid, and the vapor is drawn off by the steam jets and is condensed in a separate condenser. The cooled brine is pumped back to the supply tank from which it was drawn, and repeats the circuit.

Area of Rudders

Q.—Can you give me some information as to the areas of rudders for various speeds, lengths and types of ships, in terms of the fraction or percentage of the underwater lateral plane? The only data I have been able to find is given in Simpson's, and is not very complete. I realize, of course, that the actual area for efficient handling of the ship depends upon many factors, such as fullness, stream line flow to the rudder, number of screws, etc., but I would appreciate very much any such practical factors as you can let me have. L. A. B.

A.—Usually the area of rudder to assign to a prospective design is arrived at in no other way than by making a rather rough comparison from some vessel of similar general characteristics which has handled reasonably well in service. There is no scientific method of proportioning a rudder to give any stated turn in a definite time interval. The value given in Simpson for freighters is small, 2 percent being a more common value for moderate-sized vessels that handle well. In general, the greater the size or speed, the smaller the percentage becomes. Naval vessels of large size will have about 2 percent of the immersed lateral plane and 3 percent for moderate size, with about 2 percent for high-speed vessels of the torpedo-boat type. Since steam steering gear has become universal, a deficiency in rudder area except for very slow-speed vessels, is rarely noticed, as the rudder is quickly forced to a greater angle than should be necessary. The lower half of the rudder is more effective than the upper half.

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Connecting Rod Bearings

Q.—Will you please tell me why the bearing on the crankpin end of the connecting rod is babbitted and the one on the wrist-pin end is not, when they both have to carry the same load? B. B.

A.—The proper area for bearings is obtained by considering both the pressure per square inch and the rubbing speed. If the rubbing speed is low the pressure may be high (1,000 pounds per square inch at wrist-pin end), and if the speed is high the allowable pressure must be smaller (500 pounds per square inch at crank-pin end). For high pressures and low rubbing speeds, as at the wrist-pin bearing, babbit is undesirable.

Remedies for Sparking of Brushes

Q.—Recently the brushes on our generator (D. C. 220 volts) have given a good deal of trouble through sparking, and, so far as I know, no change has been made in the instrument. Can you suggest what is the probable trouble and a remedy for it? E. M.

A.—The trouble may be due to inaccurate brush setting, which can be remedied by (1) grinding the brushes with coarse and then fine sandpaper (not emery) to a perfect fit, and then (2) making the interval between brush sets exactly equal. This latter had best be done by actually measuring the distance between brush sets by a steel tape. The brush pressure should be adjusted to a "happy medium," too little pressure being evidenced by chattering of the brushes and too great pressure by heating of the commutator.

Application of the Theory of Mechanical Similitude

Q.—With a boat model 10.5 inches long and 2.5 inches beam, with an area of 72 square inches on the bottom, nine ounces will cause a difference of .75 inch on the waterline. What weight is required to cause a difference of 12 inches on the waterline in a boat built from this model 21 feet long and 6 feet beam, with a bottom area of 72 square feet? H. E. N.

A.—For problems of this sort there are two things to bear in mind. *First,* that we must deal with similar shapes; that is, the full-sized boat must be exactly the same number of times greater than the model, in length, breadth, draft and all linear dimensions; *second,* that, while the linear dimensions (length, breadth, draft, etc.) go up in this proportion, the area of surfaces goes up proportional to this scale squared (multiplied by itself), while the displacement and volume go up proportional to this scale cubed (multiplied by itself twice). For example, if we have a small cube 12 inches on a side and we wish to consider one 12 feet long, *all* dimensions must be 12 times as great and the volume of the large cube will be $12 \times 12 \times 12$ as great. In the same way, if a rectangular solid 3 inches broad by 10 inches long by 2 inches deep is to be increased to one 24 times as large, *every* linear dimension goes up in that proportion, and the large solid becomes 6 feet by 20 feet by 4 feet, and its volume is $24^3 = 13{,}824$ times as great. This same conception holds for a boat. From the dimensions of your model you cannot get all the

dimensions quoted for the full-sized boat by multiplying by *one* factor of magnification, and so conclusions for a boat of these dimensions cannot be drawn from this model. If we were to have a boat exactly similar to the model and 21 feet long it would be 24 times as large and would have to have a beam of $24 \times 2.5 = 60.0$ inches or 5 feet, instead of the 6 feet quoted. For a boat of this sort, which is similar to the model, it would take $9 \times 24^3 = 12{,}400$ ounces, or 7,780 pounds, to increase the waterline $\frac{3}{4} \times 24 = 18$ inches.

Length and Breadth to be Used for Hull Coefficients

Q.—In calculating the following molded coefficients what is the proper length and breadth to use and the reason for that choice? The coefficients are, the block, midship section, prismatic and water plane. The coefficients are to be computed for a distance below and above the normal load waterline. The midship section may have a flare, or it may have a "tumble home" beginning below the normal waterline; the stem may be straight or it may be of the clipper-bow type; the stern may extend beyond the after perpendicular as in the *Lusitania*.

With the exception of Biles' "Construction and Design of Ships" I can find no information on the above subject. Most books refer to the normal load waterline condition, but Biles says to use the length between perpendiculars and the molded beam, keeping these dimensions constant. C. A. F.

A.—The selection of the proper length presents more difficulty than beam and draft and no fixed rule can be given which will hold for all types. In general the length to be used for block, prismatic and water line coefficients should be the one which gives the nearest approximation to the *extreme length* of the *hull proper* (exclusive of rudder, propeller aperture, etc.) when in *normal loaded condition.* For immersions greater or less than this no other length should be used, as usually the actual changes in length due to varying immersion affect the resultant coefficients but slightly and the coefficients for the other immersions are of less importance.

The extreme perpendiculars arbitrarily accepted by the various shipyards usually serve as the limits for measurement and may diverge appreciably from that suggested above. It is worth while, however, to bear in mind that the displacement determination is rarely accurate closer than 2 percent, so that great refinement is unnecessary. For example, the error injected into the block and other coefficients involving length by taking the length to the rudder-post instead of to the stern-post of a steam vessel with a propeller aperture coming well above the load water line might be for a 400-foot ship about 6 feet, or 1½ percent. If the length be taken as first suggested it has the further advantage of being the proper length to use in resistance calculations.

The beam used should be the *extreme molded beam* of the hull *below the normal load waterline.* If there is an excessive flare beginning below the load waterline judgment must be used, but usually the molded beam *at the waterline* should be selected for this case.

For a vessel with a level keel line the *extreme molded draft* should be used, and for keels with a drag the *mean draft,* except in the case of the midship section coefficient, which should be determined from the extreme molded draft *at the midship section.*

Size of Low-Pressure Piston Rods

Q.—Will you kindly explain why and by how much the piston and connecting rods of low-pressure cylinders may be reduced in size? Assistant Engineer.

A.—In a triple-expansion engine it is normally desired that one-third of the work be performed by the high-pressure cylinder, one-third by the intermediate and one-third by the low. When the low-pressure cylinder is divided into two that means that the work done in each cylinder is normally one-sixth of the total indicated horsepower of the engine. In this case the rods for the low-pressure cylinders are properly of reduced size, as they are performing less work. The reduction is usually about 40 percent in diameter. With the present proportions of cylinder volumes in use with triple-expansion engines it is doubtful if any radical change is made in the distribution of work by the change of location of the cylinders, and there appears to be no theoretical reason why the rods should not be reduced, whatever the arrangement of the cylinders may be. Frequently, from the operating point of view, it is desirable to have all rods of the same external diameter, in which case, when hollow rods are used, the holes in the low-pressure rod may be of greater size than those in the high and intermediate.

NEW BOOKS FOR MARINE ENGINEERS

Johnson's Steam Vessels of the Atlantic Coast. Size, 8¼ by 4½ inches. Pages, 204. New York, 1916: Eads Johnson Publishing Company. Price, $5.

In recent years Mr. Eads Johnson, naval architect, New York, has published annually a very convenient pocket list of the steam vessels entering New York harbor. The demand for this book and its recognized value have led to the publication this year of the first edition of a similar book covering the steam vessels of the Atlantic coast. In this book the publishers have furnished to the marine trade a complete and accurate list of the mercantile steam vessels of American register on the Atlantic seaboard, together with the principal particulars of each vessel, such as date and place of construction, dimensions of hull, tonnage and types and sizes of engines and boilers. All this information is given in such form as to be convenient and instantaneously available. The lists are arranged as follows: First, list of owners; second, list of vessels with their owners; third, list of former names of vessels with present names; fourth, steamship companies and owners with their vessels in detail.

Mechanical Engineers' Pocketbook. Ninth Edition. By Wm. Kent, M. E., Sc. D. Size, 4 by 6¾ inches. Pages, 1,526. Illustrations, over 250. New York, 1916: John Wiley & Sons, Inc. Price, $5 net.

Since the eighth edition of this book was published, five years ago, there have been notable advances in many branches of engineering rendering obsolete portions of the book which at that time were in accord with practice. In addition, many engineering standards have been changed during the five-year period, necessitating a thorough revision of many sections of the book. The absolutely necessary revisions to bring the book up to date have involved changes in over 400 pages of the eighth edition and the addition of over 150 pages of new matter. The treatment of many of the subjects in the earlier edition has been condensed into smaller space to enable the insertion of the new matter without increasing the size of the book to unwieldy proportions, with the result that the new edition actually contains only 64 more pages than the previous edition.

Extensive revisions have been made in the subjects of materials, mechanics, fans and blowers, heating and ventilation, fuel, steam boilers and engines and steam turbines. The chapter on machine shop practice has been rewritten and practically doubled in size. The new matter includes additional data on planing, milling, drilling and grinding, together with an elaborate treatment of the subject of machine tool driving. The subject of electrical engineering has been completely rewritten. A valuable feature of the new edition is the addition of numerous tables, giving the latest standards for various engineering materials. Indispensable as this book has become, a copy of the new edition should find a place in every engineer's library on account of the value of the new matter which it contains.

Marine Articles in the Engineering Press

Standardization of Stability Curves—Calculations for Stability of Vessel in Damaged Condition—Propeller Mysteries

The World's Shipbuilding in 1915 (*excluding warships*).—Excluding all vessels intended for naval purposes, 327 vessels of 650,919 tons were launched in the United Kingdom in 1915. Of these vessels, 317, aggregating 648,629 tons, were steamers, and 10, aggregating 2,290 tons, were sailing vessels or barges. The output of the mercantile tonnage in the United Kingdom during 1915 shows a decrease of 1,032,634 tons, as compared with that of last year. Of the total output, nearly 85¼ percent has been built for registration in the United Kingdom. The largest vessel launched during the year was of 13,400 tons, 3 were over 10,000 tons each, and 33 over 6,000 tons each. The average tonnage of steamers launched in the United Kingdom during 1915 is 2,046 tons, but, if steamers of less than 500 tons are excluded, the average of the remaining steamers reaches 3,791 tons, which is a considerable reduction as compared with the mean of the averages of the five previous years. Six steamers were launched, with a total tonnage of 53,192, which will have Parsons geared steam turbine engines. Two vessels, with a total of 10,700 tons, were fitted with internal combustion engines. Seven vessels, with a gross total tonnage of 43,665, were built on the Isherwood system of longitudinal framing. Five steamers of 31,684 tons were for the carriage of oil in bulk. Eight vessels of 66,334 tons were built with cruiser sterns. The output of the leading districts was as follows: The Clyde, 205,073 tons; the Tyne, 124,001 tons; the Wear, 110,942 tons; Hartlepool, 59,308 tons; Middlebrough, 58,574 tons, and Belfast, 33,729 tons. Incomplete statistics show that outside of the United Kingdom 416 vessels of 550,719 tons were launched during the year, which represents a decrease of nearly 53 percent, as compared with the figures for 1914. 7 tables. 1,200 words.—*Lloyd's Register of Shipping,* January.

The Standardization of Stability Curves.—By Wilfrid Ayre. Apart from the primary object of investigating and diagrammatically representing the stability of a particular vessel, much information of great technical value can be obtained from curves of statical stability, provided they are reduced to a common basis or standardized in such a manner as will permit of their being readily applied to other vessels whose dimensions, proportions and form have been reduced to a standard comparative basis. It is usual to construct ordinary curves of statical stability, the ordinates of which represent values of righting arm or righting moment at various degrees of inclination. It is also customary to construct cross curves of statical stability, each showing the variation of righting arm or righting moment for fixed angles of inclination to the upright and varying displacement. Neither the ordinary curves nor the cross curves are capable of being readily applied to another vessel during, for instance, the designing stages when the exigencies of time forbid the construction of actual curves for the various range of dimensions which may be proposed. The principal object of this paper is to describe a method of standardizing statical stability curves of form which is thought to be of value in designing vessels by providing reliable means to construct curves for various dimensions and conditions proposed without incurring the construction of body plans. The second object of the paper is to illustrate the fact of systematically varying the proportion of beam and depth and of increase or decrease in the fullness or fineness of form. The author hopes that the suggestion of standardizing information already obtained, and its subsequent use as outlined in the paper, may be a means of contributing to an increased tendency to give fuller consideration to the supremely important question of the stability of ships. 22 illustrations. 3,250 words.—*Transactions of the North East Coast Institution of Engineers and Shipbuilders,* February.

The Time Element and Related Matters in Some Ship Calculations.—By Professor J. J. Welch. In estimating the stability and trim of a vessel after specific damage, attention is generally confined to the results of calculations for the particular case, where the total loss of buoyancy resulting from the damage has been compensated for by deeper immersion of the vessel, and on these results opinion is based as to whether the ship will be safe or not after such damage; but at some intermediate time before the calculated final condition is reached it may happen that other conditions supervene which render the conclusions drawn from that calculation invalid and it becomes necessary, therefore, to study intermediate conditions through which the vessel passes to insure that no factors will be introduced which will vitiate the final result. Two illustrations of this are given, floating bodies of regular form being taken for the sake of simplicity and some closely related questions are briefly discussed. The first case is that of a box-shaped vessel 200 feet long, 30 feet wide, floating at 10 feet draft and having its C. G. 11.8 feet above the keel, when a central transverse compartment 50 feet long assumed empty is opened up to the sea. An investigation is made to learn if this vessel will be stable after damage and free from danger in still water with a row of side lights open, the lower edges of which are 15 feet above the keel. The ordinary calculations for the final condition of the vessel would lead to the conclusion that the ship would be stable after damage and safe under the conditions assumed with the ports open; but it is well known that in the early states of the filling of such a compartment there will be a decided fall in the position of the virtual metacenter. In the early stages the stability will be inactive and the ship will evince a tendency to "loll," and, if the hole through which the water enters is of comparatively small dimensions there would be an appreciable time for the list to develop. The investigation of this action shows that at some intermediate time the ship will attain a position in which the lower edge of the port holes are practically 9 inches below the water, and, therefore, in actual practice the vessel would be unsafe under the conditions assumed. Another case mentioned where the element of time has an influence on the final result is that of the box-shaped vessel previously mentioned, but with the C. G. 10.5 feet above the keel to insure transverse stability during flooding, to have two end compartments, each 25 feet long. An investigation is made of the results in the event of the foremost compartment being bilged by collision. The extreme case is assumed, that of the forward compartment being first opened to the sea, and then, after that has been filled, the after compartment is opened. The final calculated results will always be the same, but supposing that the inner bulkheads of the end compartments terminate 15 feet above the keel and the end compartments are not

filled uniformly, the vessel might take up a position in which the water would be several inches above the top of the inner bulkhead of the foremost compartment and would overflow into the adjacent compartments, with subsequent foundering of the vessel. Such considerations have an important bearing upon the question of providing flooding or pumping appliances to introduce water at one end of the vessel to correct trim due to damage to the other end, with a view to improving the chances of safety of the vessel. In any event, in certain classes of ship calculation problems it is not always sufficient to consider the result of the usual conclusions relating to the time when the loss of buoyancy involved by damage is finally restored, but regard must also be paid to the position and condition of the vessel at intermediate times, since factors may be introduced at such stages which render the results of the usual conclusions unattainable in practice. 6 illustrations. 4,700 words.—*Transactions North East Coast Institution of Engineers and Shipbuilders,* January.

The Mystery of the Screw Propeller.—By Captain C. W. Dyson, U. S. N. For many years after the adoption of the screw propeller for the propulsion of ships the seeming vagaries in its performances in actual service of propulsion have cast a mystery over it and over the laws governing its action. The greater part of this mystery is, however, not due to the propellers, but can be directly laid upon the carelessness with which the trials of ships have been run and data collected, while the greater part of the remainder can be attributed to the effect of variation in hull form in interfering with the proper flow of water to the propeller, thus seriously decreasing the propulsive efficiency; in producing more or less wake and so adding to the propulsive efficiency; in incorrect estimate of effective horsepowers required for given speeds, basing these powers on estimates of frictional and residual resistances with an entire neglect of the influence of appendages, such as struts, bilge keels, etc. The small remaining portion of the mystery may be ascribed to the propeller itself and is due to the myriad variations in blade form and section which have been used, these appearing to depend upon the taste of the designer and upon the lack of a consistent basis of comparison for the analysis of the propeller performances. As time passed, the model tank by which a more nearly accurate value of the effective horsepower may be obtained became available, also more accurate instruments for the measurement of indicated and shaft horsepower, better mechanical construction of propelling engines by which frictional losses have been greatly reduced and brought to a more nearly competent and even value; more care in running trials over the measured mile, combined with a better knowledge of the effect of shallow water on the course, both tending toward the production of better data, and these improvements have resulted in the production of data from which curves of analysis may be drawn, rendering it possible to take any propeller which may be proposed for a given vessel and to estimate a curve of performance for any condition with more than a fair degree of accuracy. The comprehensive discussion which the author gives, together with rules and examples as a partial explanation of the mystery of the screw propeller, shows that the efficiency of the performance of the propeller is seriously affected by its position in relation to the hull of the vessel which it is driving and by the fullness of the hull lines at the after body; that for any given propeller working under constant hull conditions of form the effective horsepower for any given engine power remains constant, and this is independent of the speed of the vessel; that for any given propeller, working under constant hull conditions of form, the revolutions necessary to deliver any given effective horsepower vary with the speed of ship, the engine power remaining constant; that model tank effective horsepower curves are correct and that the appendage resistance varies according to the "law of comparison"; that trials of vessels over shallow water courses should be prohibited, as such courses vary entirely the character of the water flow to the propellers; that deep water trials, by the close agreement of actual with estimated results, demonstrate the correctness of the above statement; that the large estimate of from 3 to 5 percent error usually accredited to torsion meters and indicators used on official trials is incorrect, particularly at high power; that results obtained from model tank experiments with model propellers will be correct when the model is properly proportioned to the full-sized propeller, which does not mean that the fixed ratio of the propeller and of the model shall be the same, and the chart condition of the model propeller working with no hull, or rather behind a phantom ship can be calculated, and, finally, that where vessels are entered into competition based upon relative efficiencies of their propelling machinery, it is a positive error to base these efficiencies upon speed or revolutions, as both speed and revolutions for equal loads upon the propellers represented by equal effective horsepowers vary with the loading of the hull, foulness of ship's bottom, weather and sea, while the engine power for this effective horsepower remains constant, except in cases where propellers are foul or their blades distorted. It shows, therefore, that the true criterion for efficiency of propelling machinery should be indicated or shaft horsepower developed by that machinery and that the speed of ships and revolutions of propellers should be neglected. 5 illustrations. 20,000 words.—*Journal of the American Society of Naval Engineers,* November.

Turbine Machinery of the Steamship Bingera.—A highly interesting piece of work was recently carried out by Messrs. McKie & Baxter, of the Copland Works, Govan, consisting of the replacement of the turbines of the steamship *Bingera* with triple expansion reciprocating engines. The *Bingera* is a vessel of 2,092 gross tons, 300 feet 3 inches long, built in 1905 for the passenger and cargo trade of the Australian coast. The original propelling machinery, consisting of a three-shaft arrangement of steam turbines designed to drive the vessel at a speed of from 16 to 17 knots, proved unsatisfactory. The owners therefore decided to remove the turbines and fit three sets of reciprocating engines, operating at a lower speed than was necessary for the economical operation of the turbines. The reciprocating engines have cylinders 15 inches, 24 inches, 27 inches and 27 inches in diameter by 21 inches stroke, running in ordinary service at 225 revolutions per minute, developing 3,000 indicated horsepower. The vessel is now in service and is giving excellent results, a special feature being the entire absence of vibration with the four-crank triple expansion engines. The alteration seems to be justified by an increase of speed, together with a saving of coal. 10 illustrations. 600 words.—*Engineering,* December 17.

Men of the American Merchant Marine.—Returns to the Bureau of Navigation, Department of Commerce, show that on June 30, 1915, the 26,701 merchant vessels of the United States documented for foreign or domestic trade were manned by 152,133 officers and men, excluding masters. Including the masters, and also the men and officers manning yachts, there is a grand total of 187,921 officers and men for 28,257 documented merchant vessels and yachts, aggregating 8,487,331 gross tons.

Shipbuilding and General Marine News

Contracts for New Ships—Marine Terminal Improvements—Recent Launchings—Improved Appliances—Personal Items

One of the most important announcements made during the past month was that of the completion of plans for the complete reorganization of the American Bureau of Shipping with a view to placing this classification society upon a substantial foundation both technically and financially. This reorganization has the enthusiastic support of all classes interested, particularly shipowners, shipbuilders, naval architects and underwriters, who have all joined together to make the American Bureau a classification society having a reputation that will command the respect of the world.

American maritime interests are to be congratulated upon the election of Mr. Stevenson Taylor, president of the Society of Naval Architects and Marine Engineers, as president of the Bureau. Mr. Taylor's acceptance of this post is ample assurance that American shipping interests will be thoroughly protected by an efficient and progressive American classification society conducted on broad administrative and scientific lines. The Bureau is further strengthened by the election of Mr. Antonio C. Passano, president of the Great Lakes Engineering Works, as first vice-president and Mr. Frank G. Macomber, a well-known marine underwriter, of Boston, as second vice-president. The former secretary of the organization, Mr. John W. Cantillion, has been elected secretary and treasurer.

In addition to his duties as president, Mr. Stevenson Taylor has temporarily assumed the duties of supervisor of the Bureau, that office having been vacated by its former incumbent.

Contracts for New Ships

The American Shipbuilding Company, Cleveland, Ohio, has received a contract from the Mallory Steamship Company, New York, for four steamers of Welland Canal size, to cost about $1,300,000 (£267,000), and a contract from the Standard Oil Company, New York, for an oil tanker.

Including the above, the American Shipbuilding Company now has twenty-six ships ordered or under construction.

The Great Lakes Engineering Works, Detroit, Mich., has received a contract from M. A. Hanna & Company, Cleveland, Ohio, to build a 12,000-ton ore carrier.

The New York Shipbuilding Company, Camden, N. J., has contracts for two ships for parties whose names have not been made public.

The Seattle Construction & Dry Dock Company, Seattle, Wash., has a contract from Norwegian interests for the construction of two steel freight steamships of 7,500 tons deadweight capacity.

The Manitowoc Shipbuilding Company, Manitowoc, Wis., has received a contract from the Bay State Fishing Company, Boston, Mass., to build three power trawlers of 300 tons each.

The Gas Engine & Power Company & Chas. L. Seabury & Company, Consolidated, Morris Heights, N. Y., is building a 52-foot motor yacht for A. Selwyn, Rowayton, Conn.

Bowes & Mower, Philadelphia, Pa., have designed a 78-foot electrically driven yacht for O. J. Dale, Barengat, N. J.

The Robins Dry Dock & Repair Company, Erie Basin, Brooklyn, N. Y., will make extensive alterations to the steamship *Porto Rico,* now owned by the Bluefields Fruit Company, Bluefields, Nicaragua.

The Buffalo Dry Dock Company, Buffalo, N. Y., will cut in two the steel steamer *H. H. Brown,* preparatory to fitting her out for salt-water navigation.

It is reported from Portland, Ore., that the Willamette Iron & Steel Works and the Northwest Steel Company will co-operate to build a shipbuilding plant in Portland to cost about $1,500,000 (£308,000), and that orders have already been practically closed to build two steamships for Norwegian interests.

New Company for Designing Efficient Screw Propellers

A new company, known as the American Screw Propeller Company, with offices at 1520 Sansom street, Philadelphia, Pa., has just been formed for the purpose of designing efficient screw propellers for any type of seagoing or air craft. The method of design which this company will use is the Dyson method and Captain Charles W. Dyson, U. S. N., the inventor of this method, will be one of the consulting engineers of the company, together with Luther D. Lovekin, M. E., chief engineer of the New York Shipbuilding Company, as his associate, and Rear Admiral A. B. Willets, late director of the United States Navy Yards, as director.

The company will undertake to furnish designs for the best possible propeller for any vessel whose owner or builder can provide the necessary hull data. The business of the company will be international and correspondence is invited from all interested in propeller economy and efficiency.

New Shipyard on the Delaware River

The Pennsylvania Shipbuilding Company, recently organized with a capital of $600,000 (£123,000) and with offices in the Land Title building, Philadelphia, Pa., has purchased a site with an 800-foot waterfront at Gloucester, N. J., for erecting a new shipyard. Contracts have already been secured by the new company for the construction of six freight steamers each about 350 feet long of 12 knots speed to be driven by geared turbines. Contracts for steel plates for this work are being made for delivery at the rate of 600 tons a month, and it is expected that two of these ships will be ready for delivery within eighteen months after the yard is opened. The company already has sufficient work to keep it fully occupied for at least two and a half years.

The officers of the company are H. E. Norbom, of Philadelphia, president; Charles H. Moyer, of Philadelphia and New York, vice-president, and George S. Hoell, secretary-treasurer. Henry Lysholm, for many years superintendent of the hull department of the New York Shipbuilding Company, has been appointed general manager. About 450 men will be employed in the new plant.

Plans for the new yard provide for a basin 120 feet wide and 800 feet long, with building berths on each side arranged for side launching so that two vessels can be laid down on each side of the basin at the same time. Later it is expected to construct another basin alongside the first one, doubling the capacity of the yard. Plate and machine shops will be erected at the end of the basin,

together with a complete crane system for conveying material and for serving the building berths. Electric power supplied by the municipal power plant will be used throughout the plant, individual motors being provided for the machine tools in the plate shop and group drive in the machine shop.

At present the plant will be devoted to hull work only, contracts for the machinery for the vessels built being sublet to outside manufacturers.

Inspection of the Twin Screw Werkspoor Diesel-Engined Tank Ship Ares

During the recent visit to New York harbor of the Dutch motor tank ship *Ares,* built last year by the Netherlands Shipbuilding Company, Amsterdam, and fitted with two sets of six-cylinder Werkspoor Diesel engines, developing a total of 1,650 brake horsepower, Mr. T. Orchard Lisle, American representative of the Werkspoor Works, builders of the engines, conducted a party of prominent ship owners, shipbuilders and engineers on a tour of inspection of the vessel.

The *Ares* is 347 feet long, 46 feet beam, 27 feet 6 inches molded depth, with a draft of 22 feet, a displacement of 7,720 tons and a deadweight capacity of 5,050 tons. The vessel was designed for a service speed of 10¼ knots, but on trial developed a speed of 11¼ knots. The engines, which weigh 96 tons each, have cylinders 20½ inches diameter by 35½ inches stroke and operate on the four-cycle principle at from 115 to 125 revolutions per minute. The loaded fuel consumption of the vessel, including auxiliaries at sea, has proved to be not over 7 tons per 24-hour day, the coal consumption per indicated horsepower hour being from .316 to .326 pound. The vessel has a bunker capacity of 692 tons, giving a cruising radius on one load of fuel of 22,800 nautical miles.

It will be remembered that two of the largest shipyards in the United States, the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., and the New York Shipbuilding Company, Camden, N. J., have obtained licenses to build this type of Diesel engine in this country.

Interesting Transfer of Vessel Property

A strong indication of the urgent necessity for additional tonnage to take care of the immense shipments to be handled may be seen in the recent action of the Standard Oil Company in purchasing from the Smith Shipping Company a fleet of six freight vessels now under construction from the designs and under the superintendence of Messrs. Cox & Stevens, New York, at the yards of the Toledo Shipbuilding Company, Toledo, Ohio.

In August, 1915, the Smith Shipping Company, anticipating an unusual demand for vessels, placed through their naval architects, Messrs. Cox & Stevens, an order with the Toledo Shipbuilding Company for six vessels to carry 3,000 tons deadweight, to be propelled by heavy oil engines and to have in addition a sail plan capable of driving the vessels at a satisfactory speed under ordinary conditions at sea.

This fleet is now well under construction, and will be completed and ready for delivery shortly after the opening of navigation on the Lakes.

While it is not generally known what price was paid by the Standard Oil Company for acquiring these vessels, it may be safely said that the Smith Shipping Company would not have parted with them under present conditions without receiving an enormous profit over their investment, for the reason that the present earning capacity of these ships is so great that they will probably be able to make for their owners the entire first cost within less than one year of service.

The Standard Oil Company has the reputation of being very farsighted in all their business arrangements, and the fact that they have found it advisable and necessary to purchase these vessels is a clear indication of the great shortage of tonnage existing, and shows that even the prevailing high rates asked for available vessels will not deter operators from acquiring them.

New Steamship Companies

The Kentucky-Indiana Packet Company has been organized, with a capital of $10,000 (£2,050). The incorporators are F. N. Spellman, B. M. Green, Stephen Green and H. C. Tiedeman, all of New Albany, Ind.

The Pan-American Petroleum & Transport Company has been organized in Delaware, with an authorized capital stock of $150,000,000 (£30,750,000). The new corporation is a consolidation of several companies, among them being the Mexican Petroleum Company, New York. It is understood that E. L. Doheny, president of the Mexican Petroleum Company, will act as president of the Pan-American Petroleum & Transport Company. It is reported that the new company has already asked for bids for several oil tank steamships.

The Interlake Steamship Company, Cleveland, Ohio, controlled by Pickands Mather & Company, Cleveland, has purchased thirteen vessels belonging to the Cleveland Steamship Company. The Interlake Steamship Company's fleet now comprises fifty-two boats, and it is reported that the company will build several additional ships in the near future.

W. K. Greenbaum, president of the Indiana Transportation Company, Chicago, Ill., is considering establishing a regular steamship service between Atlantic City, N. J., and New York.

The Southern Transportation Company has been incorporated to engage in water transportation. The incorporators are H. Spielberg, S. Ranch and B. Bernstein, 100 Clark street, Jersey City, N. J.

Marine Terminal Construction and Improvements

The C. Reiss Coal Company, Superior, Wis., will soon award a contract for building a dock to cost about $300,000 (£61,500).

S. W. Bowen, chief designing engineer, St. Louis, Mo., has drawn plans for establishing a municipal dock and warehouse at the foot of North Market street, at an estimated cost of $255,000 (£52,250).

The Canadian Locomotive Company, Kingston, Ont., Canada, has applied for permission to build a 400-foot pier in Kingston Harbor.

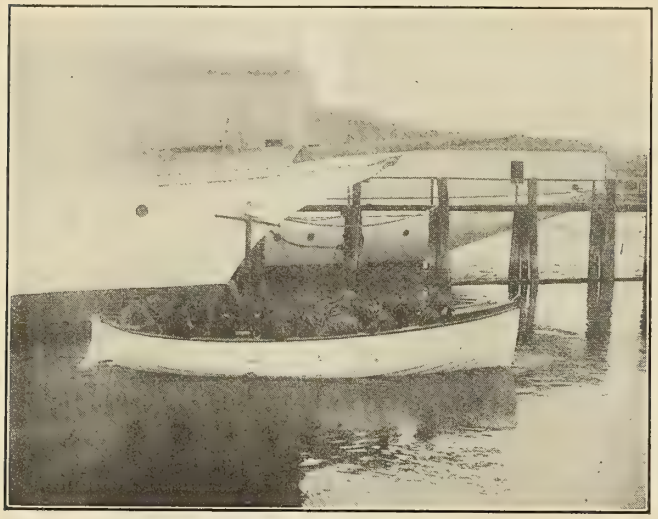
The Sinking Fund Commission of the city of New York has adopted the plans of Dock Commissioner R. A. C. Smith, Pier A, North River, New York, for the improvement of the water front on the East River between State and Orange streets, Brooklyn. The plan is for eight piers varying in length from 585 to 660 feet, to be built by the New York Dock Company, State and Whitehall streets, New York.

Aimsworth & Dunn, White Building, Seattle, Wash., are preparing plans for the reconstruction of Pier 14, Seattle, at an estimated cost of $30,000 (£6,150).

The General Chemical Company, New York, has awarded a contract to John Monks & Son, New York, to build a pier at Marcus Hook, Pa., to cost about $200,000.

Motor Lifeboats

The illustration shows a 24-foot motor lifeboat, as built by the Welin Marine Equipment Company, Long Island City, New York, in both wood and steel, in compliance with the general rules and regulations of the Board of Supervising Inspectors of the United States Steamboat Inspection Service. These boats have a beam of 7 feet 3 inches and a draft of 3 feet 3 inches. The motor, a 2-cylinder, 4-cycle, 4½- by 5-inch Speedway engine, is



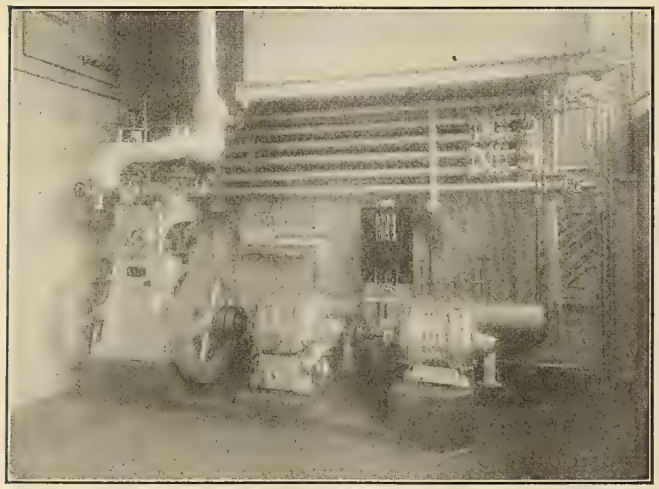
24-Foot Motor Lifeboat

installed in a thoroughly protected watertight inclosure. A 40-gallon gasoline (petrol) tank is installed under the forward deck, while galvanized iron air tanks are installed under the seats and the after deck. A water tank of 15 gallons capacity is placed forward of the engine.

The steel launches are constructed of No. 16 B. W. G. steel, the keel, stem and stern post being of flat steel. The wood launches are of lap strake construction, having ½-inch cypress planking, and the frames and keel of oak. Lifting tackles are arranged at the bow and stern.

A Turbo-Gear Installation

The illustration shows an installation of the "internal herringbone" gear reducer manufactured by the Turbo-Gear Company, Baltimore, Md., connecting a Frick Company ammonia compressor with an electric motor running at 1,800 revolutions per minute, the speed reduction being



Turbo-Gear Installation

13 to 1. An efficiency of 98 to 99 percent is guaranteed at full load on any turbo-gear having a single train of gears. With the single train of gears, speed ratios as high as 30 to 1 can be obtained. The chief reason for the superior qualities of this gear, it is claimed, is the combination of two ideal conditions in gearing—namely, that of an internal gear with its long arc of contact with corresponding easy approach and recession, and that of the helical gear with its continuity of action. The turbo-gear was primarily developed for use in connection with steam turbines, centrifugal pumps and kindred machinery, but a much larger field of application has been found in connection with electric motors, as illustrated in this case. The advantages claimed for this mode of power transmission include savings of from 20 to 25 percent in power costs over other drives, less floor space, elimination of strains on bearings of the motor or prime mover, since the driving and driven shafts are in one line, and lower initial costs. These transmissions are manufactured in sizes from 5 up to 1,000 horsepower.

Skandia Marine Oil Engines

The Skandia marine oil engine, manufactured by Messrs. Lysekils Mek. Verstads Aktiebolag, Lysekil, Sweden, has been placed on the market in the United States by the Marine Oil Engine Company, Inc., 30 Church street, New York. The Skandia is a two-cycle hot-bulb engine, which runs on heavy crude or residue fuel oil. It has been widely used throughout the world during the last eighteen

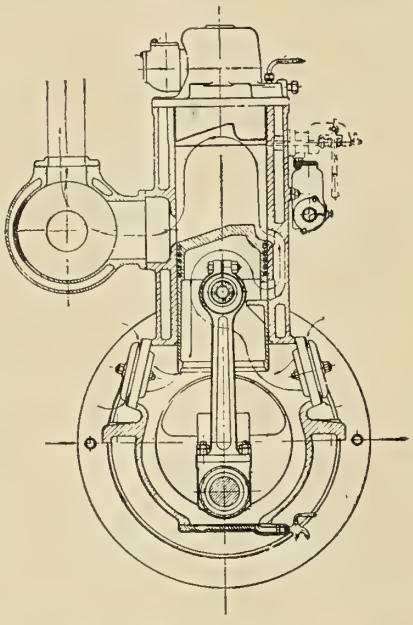


Fig. 1.—Section Through Cylinder of *Skandia* Engine

years, over 250,000 horsepower of these engines having been put on the market during that time, covering installations in tugboats, auxiliary schooners, small freight and passenger boats, fishing craft, revenue cutters, pilot boats and various government vessels. The simple construction, remarkably low consumption of fuel and the small attention required for operation (engines of this type having frequently been put in the hands of persons who have no knowledge of machinery and operated successfully) make the Skandia a most popular type of marine oil engine for work boats.

Three types of reversible Skandia engines are built. In the direct reversible engine (Fig. 2), which is built in sizes of from 60 to 500 horsepower, the reversing and

starting is effected by means of air produced by an air compressor. The maneuvering from full speed ahead to full speed astern requires only two manipulations. On each cylinder there is a maneuvering valve and a cock and on the lay shaft are cams for driving each valve ahead and astern. By drawing back a single handle all the relief cocks are opened at the same time. The cams are controlled by bringing the lay shaft forwards or backwards by means of a lever. The compressor for supplying air for starting and reversing is mounted, together with the circulating pumps and bilge pumps, in a single casting

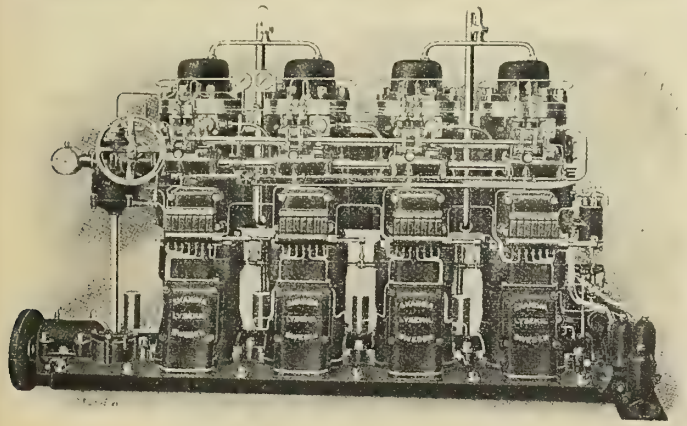


Fig. 2.—240 Horsepower Direct-Reversible Skandia Engine

forward of the first main bearing, and is driven by a special patent drive.

In the smaller engines reversing is accomplished either by a special reversing gear or by reversible propeller blades.

All Skandia engines are supplied with a governor in direct connection with which is the fuel pump worked through the medium of a cam. By means of an adjusting screw combined with the governor, the fuel feed may be regulated instantaneously while the engine is running and after the stroke of the pump has been adjusted to suit the load on the engine it will run continuously with the same number of revolutions. The governor makes it possible to regulate the speed of the engine from any position in the boat. A positive and reliable lubricating system is also provided.

Tests of a 25 brake horsepower, one cylinder Skandia marine engine recently made by the Material Testing Section of the Chalmers Institute, Gothenberg, showed a fuel consumption of only .56 pound per brake horsepower hour. With larger engines, this fuel consumption is greatly reduced.

Not only has the Marine Oil Engine Company, Inc., secured the sales rights of the Skandia engine in the United States, but also the exclusive manufacturing rights, and the company is now entering upon the construction of these engines on a large scale, having already placed orders for a number of the engines. They have also made arrangements with J. H. Hansen & Co., San Francisco, to act as exclusive Pacific coast agents for the company.

Launch of the United States Tug Wando.—The United States tug *Wando* was launched at the Navy Yard, Charleston, S. C., on March 7. The sponsor was Miss Anne Bryan, daughter of Captain B. C. Bryan, U. S. N., commandant of the yard. The *Wando* is the largest vessel that has been launched at the Charleston Navy Yard, and it is expected that its construction will be followed by many other and larger vessels.

PERSONAL MENTION

Thomas E. Brown, consulting engineer, New York, has removed his office from 17 Battery Place to 18 East Forty-first street.

Oscar R. Cauchois has been appointed general agent of the Compagnie Générale Transatlantique at New York vice Mr. Paul Faguet, retired.

Captain C. A. McAllister, engineer-in-chief of the United States Coast Guard, was appointed on March 4 for another term of four years in his present position.

Benjamin G. Fernald, manager of the New York office of the Kerr Turbine Company, of Wellsville, N. Y., has been elected director and vice-president of the company.

E. P. Worden, formerly of the Fred M. Prescott Steam Pump Works, Milwaukee, Wis., has been appointed chief engineer of Henry R. Worthington & Co., Harrison, N. J.

J. H. Torney, assistant manager of the Southern Pacific Company's Atlantic Steamship Lines, has been appointed manager of R. Martens & Co., Inc., New York, one of the largest and most influential export and shipping firms in the world.

P. P. Bourne, formerly chief engineer of Blake & Knowles Steam Pump Works, East Cambridge, Mass., is again associated with the International Steam Pump Company, in connection with special engineering work at the company's office in New York.

Thomas Benson, of Baltimore, Md., formerly in charge of the floating equipment of the Pennsylvania Railroad at Baltimore, Md., has been elected president and general manager of the Standard Shipbuilding Company, New York. Wallace Downey, formerly general manager, has been appointed general superintendent.

OBITUARY

Wallace McDonald, manager of the McDonald shipyards at Mystic, Conn., died on March 15 from heart disease.

J. S. Raymond, vice-president of the Clyde and Mallory Steamship Lines, died recently at his home in Essex Falls, N. J., aged 46.

Fred T. Jane, well known as a writer on naval subjects and editor of Jane's "Fighting Ships," died suddenly at Southsea on March 8 at the age of 45.

John W. Hill, mechanical engineer and sales manager in charge of the Detroit office of the Bantam Anti-Friction Company, died very suddenly at his home in that city on February 12.

George McCrea Boyd, shipbuilder and for many years owner of shipyards in Shanghai, China, and at Nagasaki, Japan, died on February 11 at his home in Bayonne, N. J., in his eighty-sixth year.

Captain Mark S. Kirby, commander of the freight steamship *Francis Hyde*, running from New York to New Orleans, died from heart disease on February 18 at his home in Brooklyn, aged 73 years.

William Alfred Percy, an engineer and manufacturer, and formerly member of the firm of Henry R. Worthington & Co., steam pump manufacturers, died on February 16 at his home in New York at the age of 81.

SELECTED MARINE PATENTS

The publication in this column of a patent specification does not necessarily imply editorial commendation.

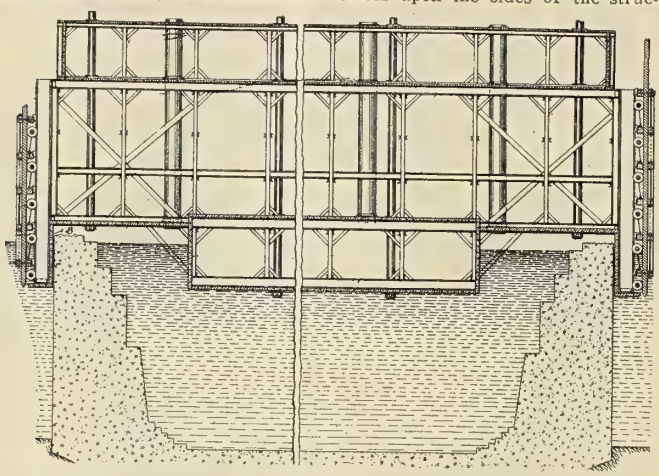
American patents, compiled by Delbert H. Decker, Esq., registered patent attorney, Millerton, N. Y.

1,146,958. BREAKWATER SHIELD FOR SUBMARINE BOATS. RICHARD H. M. ROBINSON, OF BRIDGEPORT, CONN., ASSIGNOR TO THE LAKE TORPEDO BOAT COMPANY OF MAINE, OF BRIDGEPORT, CONN., A CORPORATION OF MAINE.

Claim 1.—In a submarine or submersible boat, having a superstructure built thereon the deck of which is provided with a slot, a breakwater shield adapted to be elvated through said slot above the deck of the superstructure, and means for holding said shield in elevated position. 10 claims.

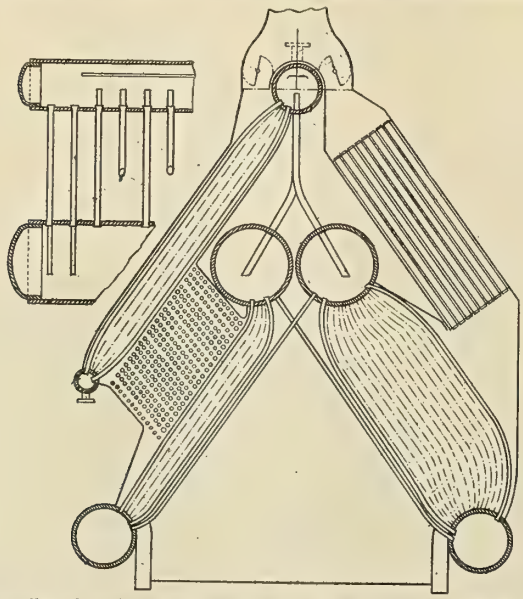
1,160,180. APPARATUS FOR CONSTRUCTING DRY-DOCKS. DUNCAN D. McBEAN, OF NEW YORK, N. Y.

Claim 1.—A structure for use in building dry-docks and like purposes consisting of a water-tight compartment capable of floating the structure and all its appurtenances, air-tight chambers along its sides open at the bottom in which the side walls may be built up from the foundation when the structure is sunk to the bottom, water compartments for loading the structure, and anti-friction devices upon the sides of the structure for running over the face of surrounding sheeting when the structure is raised or lowered. Seven claims.



1,168,748. WATERTUBE BOILER. WILHELM SCHMIDT, OF WILHELMSHÖHE, NEAR CASSEL, GERMANY, ASSIGNOR TO SCHMIDT'SCHE HEISSDAMPF-GESELLSCHAFT M. B. H., OF CASSEL-WILHELMSHÖHE, GERMANY, A CORPORATION OF GERMANY.

Claim.—In a watertube boiler, in combination with a pair of lower drums and an upper drum, two sets of watertubes connecting the lower drums with the upper drum, one set being smaller than the other set, a flue laterally of each set of watertubes, a feed-water heater in the flue adjacent to the smaller set of tubes, a steam superheater between the feed-water heater and the smaller set of tubes and in the same flue, and an air heater in the flue adjacent to the larger set of tubes.



1,170,529. SUBMARINE BOAT. HUGO GRIESHABER, OF NEW LONDON, CONN., ASSIGNOR TO ELECTRIC BOAT COMPANY, OF NEW YORK, N. Y., A CORPORATION OF NEW JERSEY.

Claim 1.—A submarine boat having a strong inner hull, a relatively weak outer hull, means for admitting water to the space between the hulls, means for admitting air to the space between the hulls to expel the water therefrom, and means for automatically regulating the pressure of the air admitted to the space between the hulls with reference to the pressure of the water surrounding the boat. Eight claims.

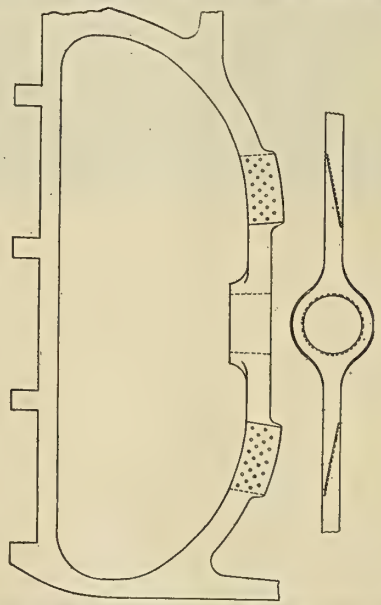
British patents compiled by G. F. Redfern & Co., chartered patent agents and engineers, 15 South street, Finsbury, E. C., and 10 Gray's Inn place, W. C., London.

22,734 OF 1914. APPLICATION OF PLASTIC COVERINGS TO METAL PLATES, SUCH AS SHIPS' DECKS. WAILES, DOVE & CO., LTD., 5, ST. NICHOLAS BUILDINGS, ST. NICHOLAS STREET, NEWCASTLE-ON-TYNE, AND C. MACDONALD.

An improved construction of flooring for the parts of ships' decks and like surfaces not exposed to weather, without the use of expanded metal embedded in the structure, which consists in coating the metal surface with a bituminous solution, securing a layer of bituminous enamel upon said coating, moulding the plastic covering upon it in sections separated one from another by open channels, and subsequently entirely filling the said channels with bituminous enamel or other elastic material for the purpose of preventing the rising or buckling of the rigid flooring.

24,118 OF 1914. CONSTRUCTION OF SHIP STERN FRAMES. J. S. TOMPKINSON, 29, QUEEN'S TERRACE, NEWCASTLE-ON-TYNE.

A stern frame for ships, wherein the boss for propeller tail shaft is constructed separately and is provided with diametrically opposite projections by which it is joined to upper and lower parts of a discontinuous fore member of the frame characterized by the member comprising the boss and its projections being constructed of ingot steel pressed to shape under a hydraulic press and connected to the fore member by welding or by scarf joints.



19,386/14. AN AUTOMATIC DEVICE FOR RECORDING ORDERS GIVEN ON SHIPBOARD FROM THE BRIDGE TO THE ENGINE ROOM. E. HARIAT, F 70 BOULEVARD CHAVE, MARSEILLES, IN THE REPUBLIC OF FRANCE.

The invention relates to arrangements whereby all orders which are sent from the bridge of a ship to the engine room are recorded by employing the device at present employed for transmitting orders. Previously when this has been done by means of electrical circuits it has been necessary to have current flowing all the time the apparatus is in use.

23,340/14. IMPROVEMENTS IN OR RELATING TO THE DECKS OF SHIPS SO AS TO RENDER THEM APPLICABLE FOR USE AS LIFE-SAVING RAFTS. T. H. SCRIMSHAW, G. G. H. SCRIMSHAW, BOTH OF MEADOW LANE, LEEDS, YORKSHIRE, AND W. G. DAVIS, OF 215 BEESTON ROAD, LEEDS, YORKSHIRE.

The promenande or other decks of the vessel, and also the roofs of the deck houses or cabins are arranged to be divided into one or more parts and to be combined with lever apparatus whereby in case of collision or other emergency one or more portions of such decks or roofs can be converted into and used as a life-saving raft for rescuing the passengers and crew and for keeping them afloat until assistance is to hand. The platform or main portion of the raft is in square or rectangular form, and it is composed and constructed of the deck boards or the boards forming the roof of the deck cabins or deck houses.

4,701/15. AN IMPROVED COMPOSITION FOR COVERING THE OUTER SURFACE OF INSULATING MATERIAL FOR SHIPS' GALLEYS, BULKHEADS, AND THE LIKE. G. LIZIERI, OF 5, MILDMAY ROAD, NEWCASTLE-ON-TYNE.

The materials used in the composition are "isinglass soap," known as such in the trade, and which is a mixture of isinglass and dry soap, and is often used as a base on which to lay gold leaf or as a kind of size and white calcined magnesite in the approximate proportions of isinglass soap, 10 parts, and magnesite, 3 parts. The soap is dissolved in mythelated spirits of wine, and the magnesite added, and the whole thoroughly mixed together to a putty-like consistency. The outer surface of the insulating material may be roughened, but is usually covered with expanded metal or other open work to hold the insulation in place, and thus forms a key for the surface-finishing composition described above. The composition is applied in a thin layer, with a trowel or other tool, and can be moulded to any shape. When dry it is very hard, smooth and damp-proof, and does not expand or contract. It can be made in any color.